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EFFECT OF HIGH EXHAUST PRESSURES ON ENGINE PERFORMANCE AND THE AVAILABILITY OF ENERGY IN EXHAUST GASES AT CRUISING CONDITIONS

BY

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AND

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by

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Submitted in Partial Fulfillment of the Requirements for the

Degree of

Master of Science in

Aeronautical Engineering

from the

Massachusetts Institute of Technology

1946

Cambridge, Massachusetts,
3 June 1946.

Professor George W. Swett,
Secretary of the Faculty,
Massachusetts Institute of Technology,
Cambridge, Massachusetts.

Dear Sir:

A thesis entitled "The Effect of High Exhaust Pressures on Engine Performance and the Availability of Energy in Exhaust Gases at Cruising Conditions" is herewith submitted in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

Respectfully,

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ACKNOWLEDGMENTS

Taylor, C. Fayette, Massachusetts Institute of Technology.

Leary, W. A., Massachusetts Institute of Technology.

Livengood, J. C., Massachusetts Institute of Technology.

General Electric Company, Schenectady, New York.

SYMBOLS

A	Area, in square inches
B	Ratio of diameter of orifice to diameter of pipe
BHP	Brake horsepower
BL	Brake load, inches of mercury
e	Volumetric efficiency
F	F/A = fuel-air ratio
H	Orifice pressure drop, inches of water
h_i	Gage manifold pressure, inches of mercury
h_e	Gage exhaust pressure, inches of mercury
I_{ex}	Exciter current, amperes
K	Orifice coefficient
l	Stroke, in inches
L	Brake arm length, in inches
m.e.p.	Mean effective pressure, p.s.i.
BMEP	Brake mean effective pressure, p.s.i.
FMEP	Friction mean effective pressure, p.s.i.
IMEP	Indicated mean effective pressure, p.s.i.
PMEP	Pumping mean effective pressure, p.s.i.
N	Revolutions per minute
P	Pressure ahead of orifice, inches of mercury, equal to P_1 plus corrected barometric pressure.
P_1	Gage pressure ahead of orifice, inches of mercury
P_e	Exhaust pressure, inches of mercury
P_i	Inlet (manifold) pressure, inches of mercury

SYMBOLS (continued)

Roto.	Rotameter reading
S.A.	Spark advance, in degrees before top dead center
T	Temperature of air entering orifice, in deg. R.
T_i	Inlet (manifold) temperature, deg. R.
T_1, T_2, T_3	Temperature measured in calorimeter, deg. F.
T_4	Temperature in exhaust pipe, deg. F.
T_e	Exhaust temperature, deg. F, used in calculation of turbine work
W_A	Mass rate of airflow, pounds per hour
W_F	Mass rate of fuel flow, pounds per hour
W_c	Compressor work, Btu/lb. air
W_t	Turbine work, Btu/lb. air
W_e	Engine brake work, Btu/lb. air
Y	Expansion factor

SUMMARY

The combination of two types of power plants to give optimum performance is one solution to demands for ever increasing power for high speed aircraft. A compressor, engine, and turbine geared to the same shaft is one such combination. Any prediction of the performance of such a system depends upon the experimental determination of the power output of the engine.

The purpose of this investigation was to determine the output of the engine at high exhaust pressures and the energy of the exhaust gases at these pressures. Values at full power conditions were obtained at the Sloan Laboratory in May 1944. (Ref. 1.) The present investigation involved the determination of these variables at cruise conditions.

A Lycoming single cylinder engine was used for the investigation. The engine was operated at a piston speed of 2100 feet per minute, with a fuel-air ratio of .065, spark advance of 28 degrees, manifold inlet temperature of 140°F, and a valve overlap of 40 degrees. The cylinder head temperature, oil pressure, and oil temperature were maintained constant at normal values.

Exhaust pressure was varied from 30 to 60 inches of mercury in 10-inch increments, and the manifold pressure from 30 to 50 inches of mercury in the same increments. Sufficient data was recorded to plot a map of the region under investigation.

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The combination of two types of power plants to give optimum performance is one solution to demands for ever increasing power for high speed aircraft. A compressor, engine, and turbine geared to the same shaft is one such combination. Any prediction of the performance of such a system depends upon the experimental determination of the power output of the engine.

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A Lycoming single cylinder engine was used for the investigation. The engine was operated at a piston speed of 2100 feet per minute, with a fuel-air ratio of .065, spark advance of 25 degrees, manifold inlet temperature of 150°F, and a valve overlap of 40 degrees. The cylinder head temperature, oil pressure, and oil temperature were maintained constant at normal values.

Exhaust pressure was varied from 30 to 60 inches of mercury in 10-inch increments, and the manifold pressure from 30 to 50 inches of mercury in the same increments. Sufficient data was recorded to plot a map of the region under investigation.

The energy of the exhaust gases was obtained by measurement of the exhaust temperatures.

It can be concluded as a result of this investigation that:

1. In the range investigated, the mass rate of airflow decreases linearly with increase in exhaust pressure.

2. Volumetric efficiency falls off linearly with increase in exhaust pressure.

3. Brake Horsepower decreases almost linearly with increase in exhaust pressure.

4. BMEP and IMEP decrease linearly with increase in exhaust pressure.

5. Indicated Horsepower increases linearly with mass airflow.

6. Mechanical FMEP and FMEP increase linearly with increase in exhaust pressure.

7. PMEP decreases linearly and mechanical FMEP increases with increase in inlet pressure.

8. Brake specific air consumption increases with increase in exhaust pressure, the rate of increase becoming greater with larger values of exhaust pressure.

9. Exhaust temperature increases linearly with increase in exhaust pressure.

10. Optimum engine p_e/p_i for the CET system increases with altitude and occurs between the values .75 and 1.0.

11. Net mean effective pressure of the CET system is slightly higher at cruising conditions than at full power conditions.

12. Net specific fuel consumption for the CET system is lower at cruising conditions than at full power conditions.

13. The turbine output in a CET system is more likely to be limited by the turbine blading material than by the energy available in the exhaust gases.

This investigation was made at the Sloan Automotive Laboratory at Massachusetts Institute of Technology, Cambridge, Massachusetts, in April-May, 1946.

INTRODUCTION

The wartime demands of the military services for aircraft capable of operating at higher altitudes and at greater speeds stimulated extensive investigation into the development of new power plants with ever higher output. The use of different types of power plants in combination to give optimum performance is a solution that offers unevaluated possibilities. The compressor-engine-turbine system is one such combination. Any prediction of the performance of such a system depends, because of the number of variables involved, upon the experimental determination of the power output of the internal combustion engine under conditions of high exhaust pressure and a measurement of the residual energy of the exhaust gases. (Ref. 2.)

The determination of these factors at full power operation was made at the Sloan Automotive Laboratory at Massachusetts Institute of Technology in April 1944. (Ref. 1.) The present investigation is a continuation of that work in the determination of these factors at cruising conditions. This work was done in the Sloan Laboratory at Massachusetts Institute of Technology in April and May, 1946.

DESCRIPTION OF APPARATUS

The set-up and arrangement of the apparatus used in this investigation is shown by the photographs, Figs. A, B and C, and by a diagrammatic sketch in Fig. D.

The engine was a single cylinder Lycoming on a Universal crankcase. It was liquid cooled, had dual spark ignition, a compression ratio of 6, a bore of 5.25 inches, and a stroke of 6.25 inches. It had single inlet and exhaust valves with an overlap of 40 degrees. The valve timing diagram is shown in Fig. E. The engine drove, besides the dynamometer, only the valve gear, the ignition breaker points, the tachometer, and the dynamometer exciter. The engine accessories, i.e., fuel, oil and water pumps, were driven by a three phase induction motor which also served as a starter.

The power output was absorbed by a Reliance Eddy Current Dynamometer. The brake load was measured by a hydraulic scale sometimes referred to as a torque cell. Brake load readings were obtained on a mercury manometer. The load was varied by varying the field current of the exciter which was cascaded through a motor generator set to load the dynamometer.

The Speed was controlled by varying the load. It was measured closely by a standard tachometer and checked by means of a 60-cycle stroboscopic light directed on a pattern disc on the flywheel.

The spark was controlled remotely from the instrument panel. The advance was measured by the conventional type neon spark disc.

The engine inlet system consisted of a combination surge and vaporizing tank. Temperature control of the inlet air was obtained by supplying either low pressure steam or cooling water to the surge tank jacket. Intake air was supplied by laboratory compressors, the flow being controlled remotely from the control panel by a Minneapolis-Honeywell gate valve which bled to the atmosphere. The air was metered in the induction line by means of a sharp-edged orifice installed in accordance with ASME specifications. (Refs. 3 and 4.) The pressure drop across the orifice was measured on a differential water manometer. The pressure before the orifice and in the surge tank was measured by mercury manometers. The temperature at the corresponding points was measured by Bureau of Standards iron-constantan thermocouples and a Tegliabu potentiometer.

One hundred octane fuel was supplied from the laboratory system. The pressure was maintained constant by the externally driven fuel pump vented to the intake system through a back-pressure diaphragm arrangement. The rate of fuel flow was controlled by a valve on the control panel and was measured by a Fischer and Porter Stabl-Vis Rotameter.

The cooling water and oil were circulated in closed systmes by means of externally driven pumps. Temperature control

of these systems was by means of heat exchangers that could be supplied with either cooling water or low-pressure steam through valves located at the control panel.

The exhaust gases were discharged through a short coupling into an exhaust calorimeter. From the calorimeter, the exhaust gases were led to a water-jacketed surge tank where they were partially cooled and then exhausted through a Minneapolis-Honeywell gate valve which was used to control the exhaust pressure. The exhaust pressure was measured by means of a mercury manometer from a tap at the exhaust calorimeter.

The exhaust calorimeter was that designed and used by the authors of Ref. 1. A scale drawing of the calorimeter is shown in Fig. F. Essentially, it is a shell within a shell arrangement fitted with baffles to retard and mix the exhaust gases. The outer shell was provided with bleeds by means of which some of the hot gases could be passed around the inner shells in order to reduce radiation losses from the center of the calorimeter where the thermocouples were located. For this investigation the only modification of the apparatus was to increase the size of these bleeds. The valves in the bleed lines were provided to control the amount of gas flowing around the inner shells. Three thermocouples were used and consisted of porcelain shielded Chromel-Alumel junctions. The temperatures given by these couples were read on a Leeds and Northrup millivoltmeter and were converted to $^{\circ}F$ by use of

Tables in Ref. 5. A fourth thermocouple, designed and manufactured by the General Electric Company for the measurement of high temperature, high velocity gases (Ref. 6) was inserted in the cylinder exhaust stack ahead of the calorimeter.

The M.I.T. High Speed Indicator was used to obtain records of cylinder pressure versus crank angle. The M.I.T. transfer table was used to obtain pressure-volume diagrams therefrom.

A Dumont Cathode Ray Oscilloscope was used in conjunction with a Sperry Magneto striction vibration pickup to detect any possible detonation.

PROCEDURE

The information sought divided the experimental work into three natural subdivisions. First, that of measuring the brake horsepower, the mass air flow, and the corresponding volumetric efficiency; second, the determination of the IHP or IMEP and m.e.p. by means of indicator cards; and third, that of measuring the exhaust temperature, all in relation to varying inlet and exhaust pressures.

The first and third quantities were determined by one set of runs using dual ignition. The second quantity was necessarily obtained by a second series of runs because one spark plug had to be removed in order to insert an indicator plug to obtain the cylinder pressure for the indicator. All runs were made with a concentrated effort to maintain uniformity and to duplicate the running conditions of Ref. 1 except that cruising r.p.m. and fuel-air ratio were used instead of full power r.p.m. and fuel-air ratio.

All runs were made maintaining engine operating variables as near the following values as possible:

Piston speed (Cruising)	= 2100 ft./min. (2025 rpm)
F/A (Cruising)	= $0.065 \pm .0010$
T_i	= $140 \pm 1^{\circ}$F.

Adjustments were made to give

S.A.	= 28° for dual ignition
	= 35° for single plug operation (used during taking indicator cards.)

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= 35° for single plug operation (used during taking indicator cards.)

The operating temperatures and pressures were maintained within the following limits:

Oil Pressure	58 - 70 p.s.i.
Inlet Oil Temperature	150°F.
Cylinder Temperature	175 - 185°F.

The first series of runs was made under the following varying conditions:

Inlet Pressure, "Hg.	Exhaust Pressure, "Hg.
30	30
30	40
30	50
30	60
40	30
40	40
40	50
40	60
50	30
50	40
50	50
50	60

The second series of runs was made under the following varying conditions:

Inlet Pressure, "Hg.	Exhaust Pressure, "Hg.
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Several familiarization runs were made to determine the best operating procedure. The ranges of values that might

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50	40

Several familiarization runs were made to determine the best operating procedure. The ranges of values that might

be expected were determined, and calibrations, adjustments, and zero or tare readings were all determined during this series of familiarization runs.

Two complete series of record runs were made for the first set of conditions. Mass airflow, brake horsepower, and volumetric efficiency were computed from readings of the orifice manometers, thermocouples, and the brake load manometer. Exhaust temperatures were obtained from the calorimeter and exhaust stack thermocouples.

A single series of runs was made for the second set of conditions. Pressure versus crank angle curves were obtained with the M.I.T. High Speed Indicator. Both light (5 lbs/in) and heavy (150 lbs/in) springs were used in order to obtain an accurate record of the effect of exhaust pressure on the PMEP as well as its effect on the IMEP.

DISCUSSION OF RESULTS

This investigation paralleled rather closely the work described in Ref. 1. An effort was made to avoid deviation from the procedure and methods outlined therein, in order that direct comparisons could be made between results obtained for the full power conditions and those at cruising power.

In the time elapsed between the initiation of that work and the completion of this, other investigators have done extensive research on the effect of high exhaust pressures on the performance of internal combustion engines. (Ref. 7.) The results reported are more general and greater in scope than could be obtained in this limited work. Further, certain recent refinements and developments in obtaining temperature measurements of high temperature, high velocity gases have been reported, which indicate a more satisfactory means of obtaining the exhaust temperature. (Refs. 6, 8, 9.) Nevertheless, it is felt that the results obtained can be used as a basis for more extensive research in this field.

The experimental results and certain computed values obtained therefrom are shown in Table I. All results are shown by means of curves in Figs. 1 - 22. The original data sheets for all runs, pressure-crank angle diagrams and calibration curves are on file in the Sloan Automotive Laboratory at Massachusetts Institute of Technology.

The effect of varying inlet and exhaust pressures on engine performance can most easily be reviewed by noting their effect upon (1) Mass rate of airflow; (2) Volumetric efficiency; (3) Mean effective pressures; (4) Brake horsepower and brake specific fuel consumption; and (5) Exhaust temperatures.

MASS RATE OF AIRFLOW:

Fig. 1 shows the effect of varying exhaust pressure on the mass rate of airflow. It is seen that, over the range investigated, for any given inlet pressure there is an almost linear reduction of mass airflow with increasing exhaust pressure. Further, the rate of variation is constant for all inlet pressures except at 50" hg. inlet pressure and low exhaust pressure. This effect is probably due to a slight drop in volumetric efficiency at higher mass rate of airflow.

The condition of inlet pressure of 30" Hg. and exhaust pressure of 60" Hg. was unstable, since the brake load required was the minimum load which could be set on the brake. All readings taken under this condition were regarded as potentially in error, and due account was taken of this in subsequent discussion.

VOLUMETRIC EFFICIENCY:

Fig. 2 shows the variation of volumetric efficiency with exhaust pressure. As would be expected from Fig. 1, it is seen that volumetric efficiency decreases linearly with increasing

exhaust pressure except at 50" Hg. inlet pressures and low exhaust pressures. At this point the maximum volumetric efficiency had probably been approached.

In Fig. 3 a plot is shown of volumetric efficiency vs. ratio of exhaust to inlet pressure (p_e/p_i). This curve is found to agree closely with the theoretical relationship of volumetric efficiency as a function of compression ratio and ratio of exhaust to inlet pressure, given in Ref. 10. Variation from this theoretical curve is probably due to valve overlap, which appears to be slightly excessive for the piston speed used.

The effect of variation of exhaust and inlet pressure on volumetric efficiency is illustrated in its relationship to the pumping loops of indicator diagrams obtained during the second set of runs. It is seen in Fig. 14 that, at a given inlet pressure, the size of the pumping loop increases with exhaust pressure and similarly, at a given exhaust pressure, the pumping loop decreases with an increase of inlet pressure.

MEAN EFFECTIVE PRESSURES:

The effect of varying inlet and exhaust pressures on indicated and pumping mean effective pressures was obtained from an analysis of indicator cards. These cards were obtained by converting the pressure-crank angle diagrams made with the M.I.T. High Speed Indicator to pressure-volume diagrams with the aid of the M.I.T. transfer table.

The effects of varying exhaust pressure at a constant inlet pressure of 40" Hg. are shown by the superposition of heavy spring (150 lbs/in) and light spring (5 lbs/in) cards shown in Figs. 10 and 11. It is seen that the indicated mean effective pressure, as measured by the area of the pressure-volume diagram, decreases with increasing exhaust pressure. (Fig. 10.) The pumping mean effective pressures, as measured by the areas of the pumping loops, similarly follow expected trends, the areas increasing with increasing exhaust pressure. (Fig. 11.)

The effects of varying inlet pressure at a constant exhaust pressure of 40" Hg. are shown by the superposition of the heavy and light spring cards in Figs. 12 and 13. As in the case of variable exhaust pressures, values of IMEP and PMEP obtained from these cards are in agreement with those which would be expected.

These cards were all necessarily obtained with single plug ignition, because the indicator unit was placed in the hole normally occupied by the other plug, and were made at a spark advance of 35° which had previously been determined as best power spark advance for single plug operation. It should be noted that statements based on quantitative values are necessarily subject to limitations due to variations resulting from single plug operation. However, it is probable that any error is small as volumetric efficiency is essentially the same at similar conditions, as seen in Fig. 2.

The BMEP was calculated from the known power output of the engine. As mentioned above, IMEP and PMEP were obtained from the area of the indicator diagrams. In Fig. 4, the BMEP at varying inlet conditions is plotted against exhaust pressure. It is seen that the BMEP decreases linearly with increasing exhaust pressure and at approximately the same rate for each inlet pressure. The IMEP at 40" Hg. inlet pressure is shown on the same figure. The similarity of the slopes of the BMEP curves was such that construction of parallel curves through the determined points of IMEP at 30" and 50" Hg. seemed logical. That this assumed relationship between IMEP and exhaust pressure at these inlet pressures is substantially correct is shown by a study of Fig. 5, in which it is seen that the indicated horsepower varies directly with mass airflow.

Fig. 6 shows a breakdown of mean effective pressures at an inlet pressure of 40" Hg. As discussed previously, the IMEP and BMEP decrease and the PMEP increases with increasing exhaust pressure. Assuming that IMEP is the sum of BMEP, PMEP and mechanical FMEP, the curve of mechanical FMEP obtained by differences increases slowly with exhaust pressure.

Fig. 7 shows a similar set of curves in which the exhaust pressure is held constant. PMEP decreases with increasing inlet pressure, substantiating the fact that mechanical FMEP increases with increasing inlet pressures.

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Fig. 7 shows a similar set of curves in which the exhaust pressure is held constant. PMEP decreases with increasing inlet pressure, substantiating the fact that mechanical FMEP increases with increasing inlet pressure.

BRAKE HORSEPOWER AND BRAKE SPECIFIC AIR CONSUMPTION:

Curves of BHP and BSAC are included in order that direct comparisons can be made with values of these quantities obtained at full power operation on this same test engine and discussed in Ref. 1. However, since both piston speed and fuel air ratio were varied in making measurements at cruise conditions, only a qualitative comparison seems justified.

Fig. 8 shows variation of BHP with exhaust pressure. As in reference 1 for the condition at full power, there is an approximate linear decrease of BHP with increasing back pressure. Similarly, curves of BSAC vs. exhaust pressure, shown in Fig. 9, follow the same order of increasing BSAC with increasing exhaust pressure.

EXHAUST TEMPERATURES:

The determination of the energy of the exhaust gases poses a difficult problem. It can be obtained by making heat measurements using a heat exchanger or by measurement of the exhaust gas temperature. Its determination by measurement of the exhaust temperature seems the easier, but this method has certain attendant difficulties in the case of a single-cylinder engine. In this case there is a very unsteady flow of the hot exhaust gas, the temperature of which varies widely. This variation from initial opening to final closing of the exhaust valve, in the range of operation of this investigation, was computed to be about 800°F. (Ref. 10.)

The exhaust calorimeter, as described in Ref. 1 and used in this investigation, was designed as a combination surge tank and mixing chamber. It was felt that measurement of the gas temperature after being mixed and brought substantially to rest in the calorimeter would give a valid measurement of its total energy.

Ref. 1 indicates that no constant temperature could be measured but that there was a wide variation in temperatures within the calorimeter at three points, each located successively further from the exhaust valve. This variation was probably due to radiation losses because the temperatures decreased with distance from exhaust port. It was suggested in Ref. 1 that some of the hot exhaust gases be passed between the calorimeter shells, that form the calorimeter walls, to maintain the wall temperature constant and thereby minimize radiation losses. This proved unsatisfactory as is shown qualitatively in Fig. 15, in which it may be seen that indicated temperatures fell off, rather than all three thermocouples approaching a single equilibrium temperature. Though the basic idea of the calorimeter may be sound, a redesign and refinement is necessary before valid and reproducible results can be obtained by its use.

In order, however, to have some common ground for comparison with results obtained in Ref. 1, the exhaust temperatures measured in the exhaust calorimeter without bleed were used. Readings of the three thermocouples showed the

same variation with exhaust pressure. The variation of exhaust temperature with exhaust pressure for thermocouple No. 3 located nearest the exhaust valve, is shown in Fig. 16. As in Ref. 1, it is seen that there is a linear decrease of exhaust temperature with increasing exhaust pressure at each inlet pressure.

In the last series of runs a General Electric shielded thermocouple was placed in the exhaust stack ahead of the calorimeter. Though the thermocouple located here is subject to the wide theoretical temperature variation mentioned above, it is felt that the actual range of variation is much less due to the heat capacity of the engine cylinder parts, the heavy exhaust stacks used, and the radiation shields of the thermocouple. Here it was found, as may be seen from the curve (T_4) in Fig. 16, that the exhaust temperature appears to be independent of inlet pressure and increases with increasing back pressure. Temperatures in the exhaust stack were found to be several hundred degrees higher than those measured at the calorimeter thermocouples. This trend of increasing temperature with increasing exhaust pressure, as well as the quantitative values of exhaust temperatures, are in fair agreement with values obtained from computation using theoretical thermodynamic charts. (Ref. 10.)

The determination of the energy of exhaust gases to a closer approximation by temperature measurement should be in-

vestigated further. However, at present, the temperature at any exhaust pressure investigated in this work is still beyond that which any modern turbine can withstand, so the performance of the turbine in a CET system is not necessarily limited by the exhaust temperature of the internal combustion engine.

COMPRESSOR-ENGINE-TURBINE SYSTEM

While the power output of the engine proper is reduced considerably with increase of the ratio, P_e/P_i, it is possible to realize a greater overall output by use of a compressor-engine-turbine system. In such a system the engine would be operating at a relatively high P_e/P_i with considerably reduced volumetric efficiency. However, utilization of a part of the energy available in the exhaust gases through a turbine increases the overall output to such degree that a successful compressor-engine-turbine system appears wholly feasible.

In order to make direct comparison of full power conditions (Ref. 1) and cruising conditions, the total brake horsepower output of a CET system was calculated on the following assumptions:

1. The turbine, engine, and compressor are directly connected to the propeller shaft.
2. The compressor works on the mixture of vaporized fuel and air.
3. Airplane speed is 300 miles per hour, indicated airspeed, and the full effect of ram on pressure and temperature is utilized.
4. Turbine and compressor efficiencies are .70. This appears low for efficiencies of present turbines and compressors but these values are assumed in order to obtain comparison with Ref. 1.

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4. Turbine and compressor efficiencies are .70. This appears low for efficiencies of present turbines and compressors but these values are assumed in order to obtain comparison with Ref. 1.

5. The system is operating in a standard atmosphere.
6. The highest measured temperature in the calorimeter is the temperature of the gas to the turbine, and the turbine is capable of operating at these temperatures. No correction to exhaust temperature is made for variation in inlet temperature from 140°F. The variation in exhaust temperature due to this effect is considered to be within the range of error in measuring the exhaust temperature.
7. The net total power of the CET system is equal to the measured brakehorsepower of the engine minus the computed power absorbed by the compressor plus the computed power delivered by the turbine.

The variation of net total brakehorsepower with engine exhaust pressure is shown in Fig. 17, at sea level, 15,000 feet and 30,000 feet. Cruising power output appears to be about 70% of full power output (Ref. 1). In Fig. 18 the variation of net total brakehorsepower is plotted against the ratio P_e/P_i. The optimum P_e/P_i increases with increase in altitude and occurs between the values .75 and 1.0. This is consistent with results obtained in Refs. 1 and 7.

Table II shows the work per pound of air for the various components of the CET system. The power absorbed by the compressor remains constant for any one condition of inlet pressure and altitude, i.e., does not change with variation in exhaust pressure. The turbine work increases with increase

in engine exhaust pressure. It also increases with altitude in greater proportion than does the power absorbed by the compressor. At $P_1 = 50"$ Hg., $P_e = 60"$ Hg., the power delivered by the turbine, in excess of the power absorbed by the compressor, accounts for approximately 18% of the net total power. This value would be somewhat reduced in a multiple cylinder arrangement because the percentage of power absorbed by engine friction would be reduced. The multiple cylinder arrangement would result in higher total output.

A more valid evaluation of the experimental results may be realized by analyzing the CET system from the viewpoint of mean effective pressures and specific fuel consumption. Ref. 2 analyzes the full power experimental results from this viewpoint. A better comparison of full power conditions and cruising conditions may be obtained with such an analysis for cruising conditions. Calculations are made for 15,000 feet and 30,000 feet, based on the same assumptions as Ref. 2:

Compressor efficiency:	= .85
Turbine efficiency:	= .90
Temperature of gases entering turbine:	From curves in Fig. 16
Atmospheric conditions:	Standard
Ram pressure at compressor intake:	1" Hg.
Engine BMEP corrected from:	Fig. 4
Friction m.e.p.	Estimated, based on Fig. 5, Ref. 2

An inlet pressure of 40" Hg. was selected as representative of cruising conditions. Test results were corrected to a compression ratio of 6.5 and an inlet temperature of 566°R. Values for exhaust pressures to 80" Hg. were extrapolated. Methods for calculation are shown under FORMULAE AND SAMPLE CALCULATIONS.

Fig. 19 shows the component mean effective pressures and the net m.e.p. under cruising conditions at 15,000 feet and 30,000 feet. Maximum net m.e.p. occurs at an engine exhaust pressure of about 40" Hg. Turbine m.e.p. increases with altitude while BMEP decreases linearly.

Fig. 20 compares component and net mean effective pressures at full power and cruising conditions at 30,000 feet. Full power mean effective pressures were taken from Ref. 2. It was expected that engine BMEP at cruising conditions would slightly exceed the engine BMEP at full power due to increased volumetric efficiency at cruising conditions. There is considerable divergence in the two curves as exhaust pressure is increased. This may be due to volumetric efficiency characteristics, i.e., volumetric efficiency at high piston speed may be reduced at a greater rate with increasing exhaust pressure than at a lower piston speed. The cruising condition IMEP values were taken directly from values determined from the indicator card. Full power IMEP values were taken from a general curve of IMEP/inlet pressure vs. P_e/P_i. This could possibly ac-

count for the difference in slope of the BMEP curves.

Since the compressor and turbine m.e.p. were essentially the same at full power and cruising, the net m.e.p. varies about the same as engine BMEP, i.e., the net m.e.p. is somewhat higher for cruising conditions.

Net specific fuel consumption at full power and cruising at 30,000 feet are compared in Fig. 21. The net s.f.c. is significantly reduced at cruising.

The results of the temperature measurements in the cylinder exhaust (T_4, Fig. 16) indicate that energy is available in such amount that it would not be unreasonable to assume an arrangement whereby the exhaust gases are available to the turbine at all times, at temperature above that at which modern turbines can operate. Assume further a thermostatically controlled intercooler so that gases enter the turbine at 1500°F. This figure (1500°F) is based on the General Electric I-40 engine (Ref. 11) with some allowance for improved turbine blading material in the future.

Fig. 22 compares the cruising net m.e.p. of such a system with the system previously assumed at 30,000 feet. Net m.e.p. is increased a small but significant amount with increased exhaust pressure and with increased altitude. The turbine m.e.p. furnishes a larger percentage of the total net m.e.p.

CONCLUSIONS

As a result of this investigation of the effects of high exhaust pressures on the performance of an internal combustion engine, the following conclusions may be drawn:

1. In the range investigated, the mass rate of airflow decreases linearly with increase in exhaust pressure.
2. Volumetric efficiency falls off linearly with increase in exhaust pressure.
3. Brake Horsepower decreases almost linearly with increase in exhaust pressure.
4. BMEP and IMEP decrease linearly with increase in exhaust pressure.
5. Indicated Horsepower increases linearly with mass airflow.
6. Mechanical FMEP and PMEP increase linearly with increase in exhaust pressure.
7. PMEP decreases linearly and mechanical FMEP increases with increase in inlet pressure.
8. Brake specific air consumption increases with increase in exhaust pressure, the rate of increase becoming greater with larger values of exhaust pressure.
9. Exhaust temperature increases linearly with increase in exhaust pressure.

10. Optimum engine P_e/P_i for the CET system increases with altitude and occurs between the values .75 and 1.0

11. Net mean effective pressure of the CET system is slightly higher at cruising conditions than at full power conditions.

12. Net specific fuel consumption for the CET system is lower at cruising conditions than at full power conditions.

13. The turbine output in a CET system is more likely to be limited by the turbine blading material than by the energy available in the exhaust gases.

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FORMULAE AND SAMPLE CALCULATIONS

AIR METER:

$W_A = AKY\sqrt{2g\rho\Delta P}$

Orifice characteristics:

Diameter = 1.378 inches

Area = 1.490 square inches

$K = 0.617$

$Y = 1.0$

$\beta = 0.45$

$$W_A = 3600 \times 1.490 \times 0.617 \times 1.0\sqrt{\frac{2 \times 386 \times p \times 70.7}{1728 \times T \times 53.3} \times \frac{H \times 1.00 \times 62.4}{1728}}$$

$$W_A = 485\sqrt{\frac{P \times H}{T}} \quad \text{lb/hr}$$

1.00 = specific gravity of water

62.4 = weight of water, lb/cu.ft.

70.7 = conversion, "Hg to p.s.f.

53.3 = gas constant

P = pressure in front of orifice, in inches of mercury; equal to gage-pressure in front of orifice plus barometer (both in inches of mercury).

H = orifice pressure drop in inches of water.

T = temperature of air entering orifice in degrees Rankine.

VOLUMETRIC EFFICIENCY:

$$e = \frac{W_A}{n\,V\,\rho_i}$$

$$W_A = \frac{485}{60}\sqrt{\frac{P \times H}{T}} = \text{lbs air/min.}$$

$$n = \frac{2025}{2} = 1012.5 \text{ suction strokes/min.}$$

$$V = \frac{\pi}{4} \times \frac{(5.25)^2 \times 6.25}{1728} = \text{displacement volume, cu.ft.}$$

$$\rho_i = \text{inlet air density} = \frac{p_i}{T_i} \times \frac{70.7}{53.3}$$

p_i = pressure in intake manifold, "Hg.

T_i = inlet air temperature, deg. Rankine.

Then:

$$e = \frac{W_A/60}{1012.5 \times \frac{\pi}{4} \times \frac{(5.25)^2 \times 6.25}{1728} \times \frac{p_i}{T_i} \times \frac{70.7}{53.3}}$$

$$e = .0001585 \ \frac{W_A\,T_i}{p_i}$$

where W_A = Airflow, lbs/hr.

T_i = Temperature in intake manifold, deg. Rankine.

p_i = Manifold pressure, "Hg.

BRAKE HORSEPOWER:

Dynamometer arm = 21.02" = 1.752 ft.

$$HP = \frac{2\pi RNF}{33000}$$

R = Brake arm, ft.

N = rpm

F = Brake load, lbs.

Diameter of Hydraulic Piston = 2.955 inches

Area of Hydraulic Piston $= \frac{\pi}{4} \times (2.955)^2 = 6.85$ sq. in.

Force on Piston, $F = p \times a$ $\qquad p$ = psi

a = area, inches

$F = 6.85\ p$

1 psi = 2.042 "Hg

$F = \dfrac{h}{2.042} \times 6.85 = 3.35\ h$ $\qquad h$ = "Hg

$N = 2025$

Then

$$Bhp = \frac{2\pi \times 1.752 \times 2025 \times 3.35\,h}{33000}$$ h = brake load, "Hg

$$Bhp = 2.26\ h$$

<u>BMEP</u>:

$$HP = \frac{\frac{PLAN}{2}}{33000} = \frac{2\pi RNF}{33000}$$

P = BMEP, psia.

L = stroke, ft.

A = piston area, sq. in.

$$P = \frac{2\,(2\pi RNF)}{LAN} = \frac{4\pi RF}{LA}$$

$$= \frac{4\pi \times 1.752 \times 12 \times 4 \times 3.35\,h}{6.25 \times \pi \times (5.25)^2}$$

$$= 6.55\ h$$ h = brake load, "Hg

SAMPLE CALCULATION OF CET SYSTEM

From Ref. (12).

Subscript 1 = before compressor or turbine

Subscript 2 = after compressor or turbine

h_a = Enthalpy of air, Btu/lb. air

h_f = Enthalpy of fuel, Btu/lb. fuel

h_m = Enthalpy of mixture Btu/lb. mixture

h_{ma} = Enthalpy of mixture per pound of air, in Btu/lb.

h_g = Enthalpy of exhaust gases, Btu/lb. gases

f = Fuel-air ratio = .065

η_c = Compressor efficiency = .70

η_t = Turbine efficiency = .70

$h_a + f\,h_f = h_{ma_1}$

$h_{m1} = \dfrac{h_{ma1}}{1+f}$ = Enthalpy of mixture entering compressor

<u>Compressor Work</u> $= \dfrac{h_{m1} - h_{m2}}{.7}$ Btu/lb. mixture

$= \dfrac{1.06\,h_{m2} - h_{m1}}{.7}$ Btu/lb. air

where h_{m2} is obtained from Ref. (12) by means of relative pressure ratio of manifold pressure to atmospheric pressure plus ram.

<u>Engine Work</u> $= \dfrac{BHP \times 2545}{lbs\ air/hr}$ = Btu/lb. air

<u>Turbine Work</u> = $.7(h_{e1} - h_{e2})$ Btu/lb. exhaust gases

$= 1.080 \times .7(h_{e1} - h_{e2})$ Btu/lb. air

where h_{e1} is obtained from exhaust temperature and h_{e2} by relative pressure ratio of exhaust pressure to atmospheric pressure.

<u>Net System Work / lb. air</u> = <u>Engine Work - Compressor Work + Turbine Work</u>

$$\text{Net Horsepower} = \frac{\text{Net Work / lb. air} \times \text{lbs. air/hr.}}{2545}$$

<u>Example:</u>

At: 300 m.p.h., indicated air speed
15000 feet altitude
50" Hg. inlet pressure
40" Hg. exhaust pressure

$$p_a = 16.88 + \frac{\frac{1}{2}(.002378)}{70.7} \times (300 \times 1.47)^2 = 18.94 \text{ "Hg.}$$

$$T_a = 465 + \frac{(300 \times 1.47)^2}{12{,}000} = 481^\circ \text{F abs.}$$

$$h_a = 19.42$$

$$h_f = .5T_f - 375 = .5 \times 481 - 375 = -134.5$$

$$h_{m1} = 19.42 - \frac{(.065 \times 134.5)}{1.065} = \frac{10.69}{1.065} = 10.05$$

	p	h_m	pr
(1)	18.94	10.05	1.417
(2)	50	<u>43.93</u>	3.74
$h_{m2} - h_{m1}$ =		33.88	

Compressor Work = $1.065 \times \frac{33.88}{.7}$ = 51.4 Btu/lb. air.

Engine Work = $\frac{72 \times 2545}{496}$ = 369 Btu/lb. air

Turbine Work:

Exhaust Temperature: 2024°F abs.

	T	p	h_e	Pr
(1)	2024	40" Hg.	416.2	378.3
(2)		29.92	308.4	159.5

$$h_{e1} - h_{e2} = 107.8$$

Turbine Work = $1.065 \times .7 \times 107.8 = 80.2$

Net Work = $369 - 51.4 + 80.2 = 398$ Btu/lb. air

Net Power = $\frac{398 \times 496}{2545}$ = 78.1 Hp #

CALCULATION OF M.E.P. AND S.F.C. FOR CET SYSTEM

M.E.P.:

Net m.e.p. = IMEP - FMEP - CMEP + TMEP

where IMEP is corrected from Fig. 4.

FMEP is estimated, based on Fig. 5, Ref. 2.

$$\text{CMEP} = \frac{2 \text{ x compressor work per unit time}}{\text{engine displacement per unit time}}$$

$$\text{TMEP} = \frac{2 \text{ x turbine work per unit time}}{\text{engine displacement per unit time}}$$

Corrected IMEP = test IMEP [(Fig. 4) from area of indicator diagram, not including pumping cycle] x $\sqrt{\frac{600}{586} \times \frac{.526}{.513}}$ where .526 is air cycle efficiency at 6.0 compression ratio. The complete correction factor is 1.04.

No correction made on exhaust gas temperature due to increase in compression ratio.

COMPRESSOR INLET CONDITIONS (1" Hg. ram)

Altitude	P_b, "Hg.abs.	T_b, $^{\circ}R$
15,000	17.9	472
30,000	9.9	424

Engine Inlet: P_i = 40" Hg. abs.; Inlet manifold temperature, 586°R

Engine Exhaust Pressure: 30" - 80" Hg. abs.

COMPRESSOR M.E.P.:

$$\text{CMEP} = \frac{778}{144} \rho_i \bullet c_p T_b \left(\frac{P_i}{P_b \times .98}\right)^{.283} - 1 \quad \frac{1}{\eta_c}$$

where: c_p = specific heat of air at constant pressure
= .24 Btu/lb-$^{\circ}$F

ρ_i = inlet density (i = P_i/T_i x 70.7/53.3)

e = volumetric efficiency (from Fig. 2)

η_c = compressor efficiency.

The factor .98 is inserted to allow for 2% pressure loss between compressor and engine manifold.

<u>TURBINE M.E.P.</u>:

$$TMEP = \frac{778}{144} \rho_i e (h_e - h_o)(1 + F) \eta_t$$

where: h_e = enthalpy at engine exhaust pressure, p_e, and temperature T_e, Btu/lb. air

h_o = enthalpy, Btu/lb. air, after reversible adiabatic expansion from exhaust conditions, p_e and T_e, to atmospheric pressure.

F = fuel-air ratio.

η_t = Turbine efficiency.

<u>NET S.F.C.</u>:

The indicated specific fuel consumption of the engine was constant at .365 lb. fuel/ihp-hr. The net S.F.C. for the CET system was calculated as follows:

Net S.F.C. = .365 x IMEP/net m.e.p.

<u>SAMPLE CALCULATION</u>:

p_i = 40"Hg.; p_e = 40"Hg.; altitude = 15,000 feet;

c = .85; t = .90

IMEP = 194.1 x 1.04 = 202 p.s.i.

FMEP = 26 p.s.i., (estimated)

ρ_i = 1.33 x 40/586 = .0907 lbs/ft^3.

e (from Fig. 2) = .884

$$\text{CMEP} = \frac{778}{144}(.0907) \times .884 \times .24 \times 472 \left[\left(\frac{40}{17.9 \times .98}\right)^{.283} - 1 \right] \frac{1}{85}$$

$$= 19.6 \text{ p.s.i.}$$

T_e (from Fig. 16) = 1971°R

$$\text{TMEP} = \frac{778}{144}(.0907) \times .884 \times (401.5 - 296.6) \times .90 \times 1.065$$

$$= 43.4 \text{ p.s.i.}$$

Net m.e.p. = 202 - 26.0 - 19.6 + 43.4 = 199 p.s.i.

Engine BMEP = IMEP - FMEP = 202 - 26.0 = 176.0 p.s.i.

Net s.f.c. = $.365 \times \frac{202}{199} = .371$.

TABLE I

M.I.T. AERO ENGINE LABORATORY

ENGINE B.S.R. BORE 5.25" STROKE 6.25" COMPRESSION RATIO 6.0

SUMMARY OF RECORD RUNS

Piston Speed 2100 f.p.m. R.P.M 2025 F/A : .065 ± .001 S/A : 28° Ti = 140°F ± 1°F Inlet Oil Temp. = 150°F Inlet Oil Press. = 58-70 psi Cylinder Temp. : 175-185°F

REMARKS	DATE	TIME	RUN	BAR. CORR.	P_1	P_1+BAR.	H	T	ROTO.	Ti	hi	he	Wf	Wa	F/A	B.L.	BHP.	Bmep	e	T_3	T_2	T_1	T_4
	1946			"Hg.	"Hg	"Hg	"H₂O	°R	CFM	°R	"Hg gage	"Hg gage	lbs/hr	lbs/hr	-	"Hg	HP.	psia		MV/°F	MV/°F	MV/°F	MV/°F
Pi = 30 Pe = 30	3/26	1545	1	30.00	.40	30.40	5.95	542	7.30	600	0	.50	18.0	280	.0643	16.7	37.7	109.4	.887	31.8	30.8	29.8	43.4
"	4/2	1445	1	29.70	.40	30.10	6.20	547	7.65	601	0	.40	18.5	284	.0652	16.6	37.5	108.6	.909	1407	1363	1320	1920
Pi = 30 Pe = 40	4/4	1325	2	30.00	.35	30.35	5.40	543	6.70	600	0	10.10	17.15	267	.0643	14.9	33.7	97.7	.847	30.8	29.5	28.3	43.6
"	4/5	1405	1	29.70	.35	30.05	5.40	539	6.90	601	0	10.10	17.45	266	.0650	15.2	33.4	99.4	.850	1363	1308	1256	1930
Pi = 30 Pe = 50	3/26	1715	3	30.00	.50	30.50	4.80	544	6.20	600	0	19.10	16.40	251	.0654	14.1	31.8	92.3	.798	30.1	29.1	28.0	45.0
"	4/4	1345	3	30.00	.35	30.35	4.90	541	6.30	600	0	19.90	16.50	254	.0648	14.0	31.6	91.8	.805	1333	1290	1243	2007
Pi = 30 Pe = 60	3/26	1745	4	30.00	.40	30.40	4.60	544	6.15	601	.05	30.80	16.30	249	.0659	11.2	26.3	73.5	.794	28.5	27.3	26.3	—
"	4/5	1530	3	29.70	.30	30.00	4.70	540	6.00	600	0	30.10	16.10	248	.0650	10.8	24.4	70.6	.791	1265	1214	1171	—
Pi = 40 Pe = 30	4/4	1435	5	30.00	10.50	40.50	8.65	535	12.50	600	10.0	.85	25.30	393	.0644	25.6	57.8	167.8	.930	34.6	33.8	32.9	44.0
"	4/5	1400	4	29.70	10.55	40.25	8.70	534	12.60	600	10.0	.90	25.50	392	.0650	25.4	57.4	166.1	.937	1529	1494	1454	1960
Pi = 40 Pe = 40	3/27	1620	3	29.80	10.40	40.20	7.85	541	11.80	599	9.95	9.85	24.35	371	.0656	23.5	53.2	158.2	.882	34.2	33.3	32.3	45.0
"	4/4	1520	6	30.00	10.50	40.50	8.05	537	11.70	600	10.0	10.10	24.30	377	.0645	23.8	53.8	156.0	.876	1511	1472	1439	2007
Pi = 40 Pe = 50	3/27	1645	4	29.80	10.40	40.20	7.20	541	11.00	600	10.0	19.90	23.30	355	.0656	22.0	49.7	144.1	.846	33.4	32.4	31.5	45.4
"	4/4	1545	7	30.00	10.40	40.40	7.45	538	11.10	600	10.0	20.00	23.40	363	.0645	22.3	50.4	146.0	.864	1476	1433	1393	2007
Pi = 40 Pe = 60	4/4	1605	8	30.00	10.45	40.45	6.90	538	10.60	600	10.0	30.10	22.70	348	.0652	20.4	46.1	133.8	.828	33.0	32.1	31.1	46.1
"	4/5	1655	5	29.70	10.45	40.15	6.90	534	10.50	599	10.0	30.10	22.6	349	.0648	20.1	45.5	131.5	.832	1459	1420	1376	2058
Pi = 50 Pe = 30	4/1	1450	1	29.70	20.7	50.40	11.05	535	18.20	601	20.0	1.10	32.0	494	.0648	33.2	75.0	217.5	.947	35.4	34.8	34.0	45.0
"	4/4	1635	9	30.00	20.75	50.75	11.35	534	18.90	599	20.0	1.40	32.8	504	.0652	33.0	74.7	217.0	.950	1564	1538	1503	2007
Pi = 50 Pe = 40	4/1	1520	2	29.70	20.65	50.35	11.15	534	18.10	600	20.0	10.00	31.95	497	.0643	32.0	72.2	209.5	.950	354	34.8	34.0	45.0
"	4/4	1655	10	30.00	20.65	50.65	11.00	535	18.10	599	20.0	9.90	31.90	496	.0644	31.5	71.2	206.5	.942	1564	1538	1503	2007
Pi = 50 Pe = 50	4/1	1545	3	29.70	20.60	50.30	10.50	534	17.00	599	20.0	19.80	30.80	482	.0642	30.1	67.9	197.0	.917	34.8	34.0	33.3	45.4
"	4/4	1715	11	30.00	20.60	50.60	10.25	535	17.10	600	20.0	20.10	30.85	477	.0647	30.0	67.8	196.5	.908	1538	1503	1472	2027
Pi = 50 Pe = 60	4/4	1740	12	30.00	20.55	50.55	9.70	535	16.30	600	20.0	30.20	30.00	462	.0649	27.8	62.8	182.1	.879	34.2	33.6	32.8	46.2
"	4/5	1725	6	29.70	20.60	50.30	9.60	536	16.30	599	20.0	30.20	30.00	460	.0652	27.4	61.9	179.1	.876	1511	1485	1450	2062

TABLE II.

CALCULATED OUTPUT CET SYSTEM.

ALT.	P_1	P_e	T_e	W_c	W_t	W_e	W_{net}	lbs.Air/hr	$H.P._{net}$
S.L.	50	30	1564	23	0	381	358	502	70.6
S.L.	50	40	1564	23	29	369	375	496	73.0
S.L.	50	50	1538	23	49	359	385	480	72.6
S.L.	50	60	1511	23	64	344	385	461	69.6
15,000	50	30	1564	51	55	381	385	502	76.1
15,000	50	40	1564	51	80	369	398	496	78.1
15,000	50	50	1538	51	97	359	405	480	77.0
15,000	50	60	1511	51	109	344	402	461	73.4
30,000	50	30	1564	75	108	381	414	502	81.6
30,000	50	40	1564	75	129	369	423	496	82.5
30,000	50	50	1538	75	143	359	427	480	80.5
30,000	50	60	1511	75	152	344	421	461	76.1
S.L.	40	30	1529	11	0	374	363	392	55.8
S.L.	40	40	1511	11	28	364	381	374	56.0
S.L.	40	50	1476	11	48	354	391	360	55.3
S.L.	40	60	1459	11	62	335	386	348	52.8
15,000	40	30	1529	38	54	374	390	392	60.3
15,000	40	40	1511	38	78	364	404	374	59.6
15,000	40	50	1476	38	94	354	410	360	59.8
15,000	40	60	1459	38	106	335	403	348	55.4
30,000	40	30	1529	63	106	374	417	392	64.2
30,000	40	40	1511	63	126	364	427	374	62.8
30,000	40	50	1476	63	138	354	429	360	60.8
30,000	40	60	1459	63	148	335	420	348	57.4

T_e = exhaust temperature, °F, used in calculation of turbine work.

W_c = compressor work, BTU/lb.air.

W_t = turbine work, BTU/lb.air.

W_e = engine brake work, BTU/lb.air.

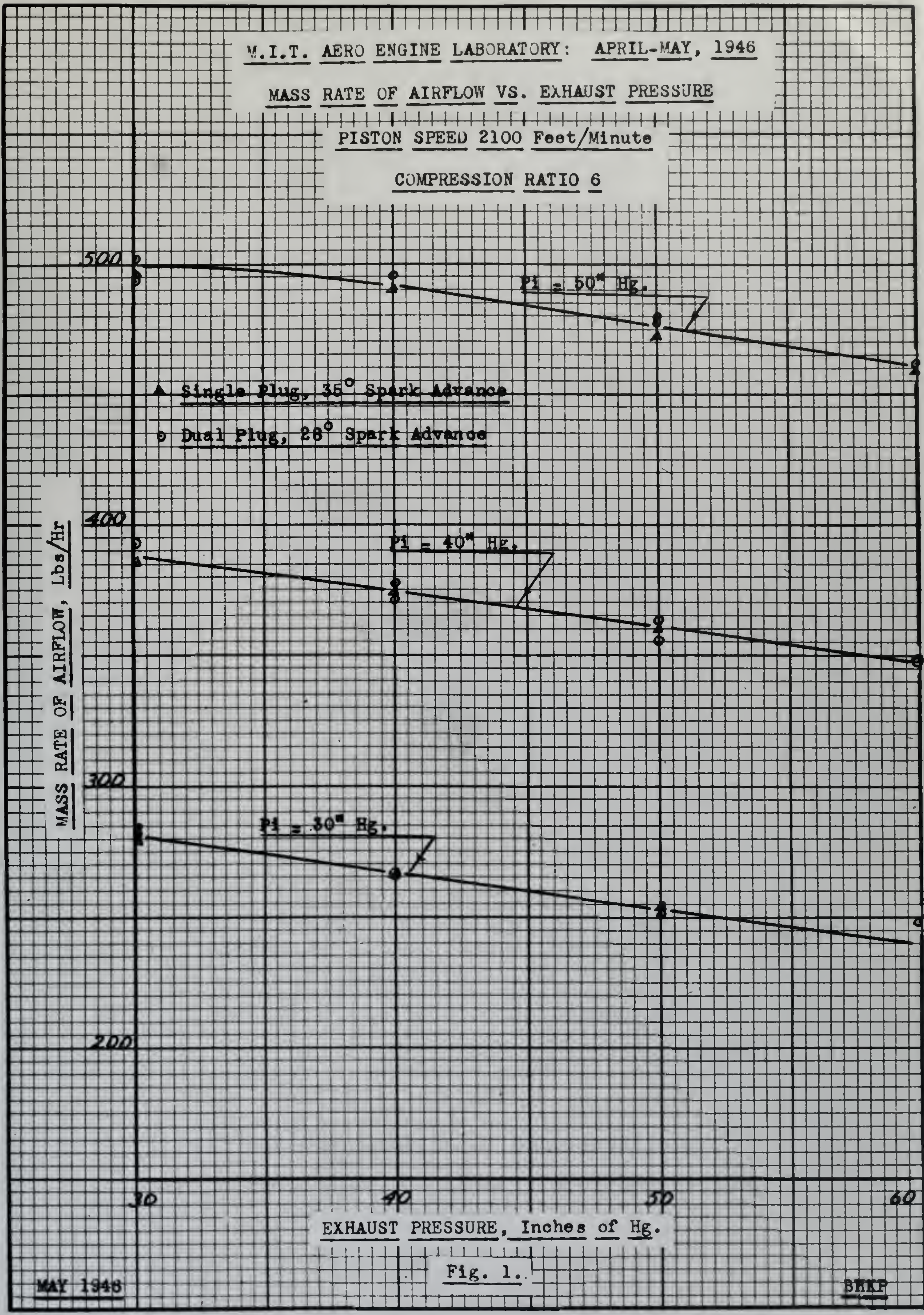
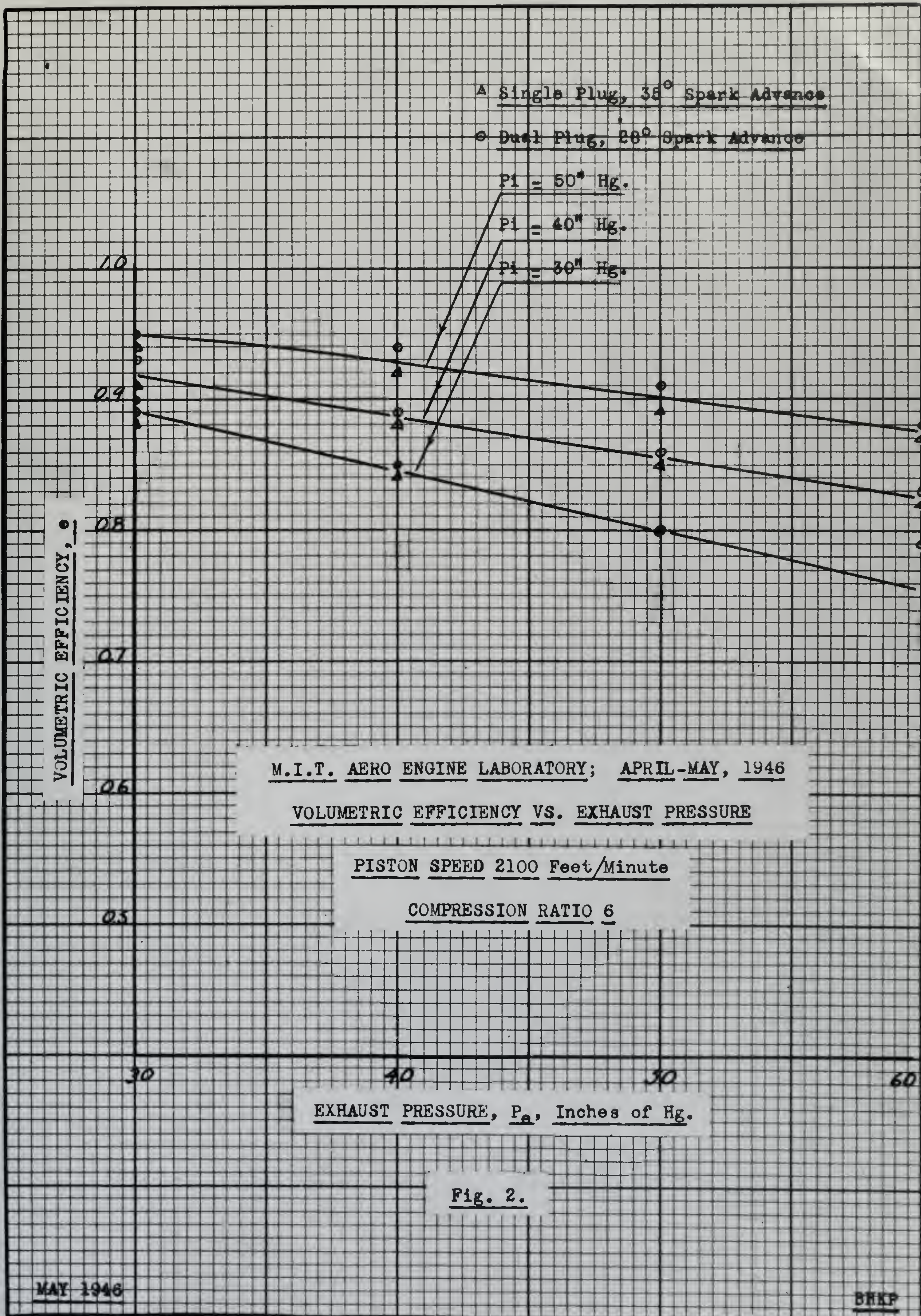


Fig. 1.

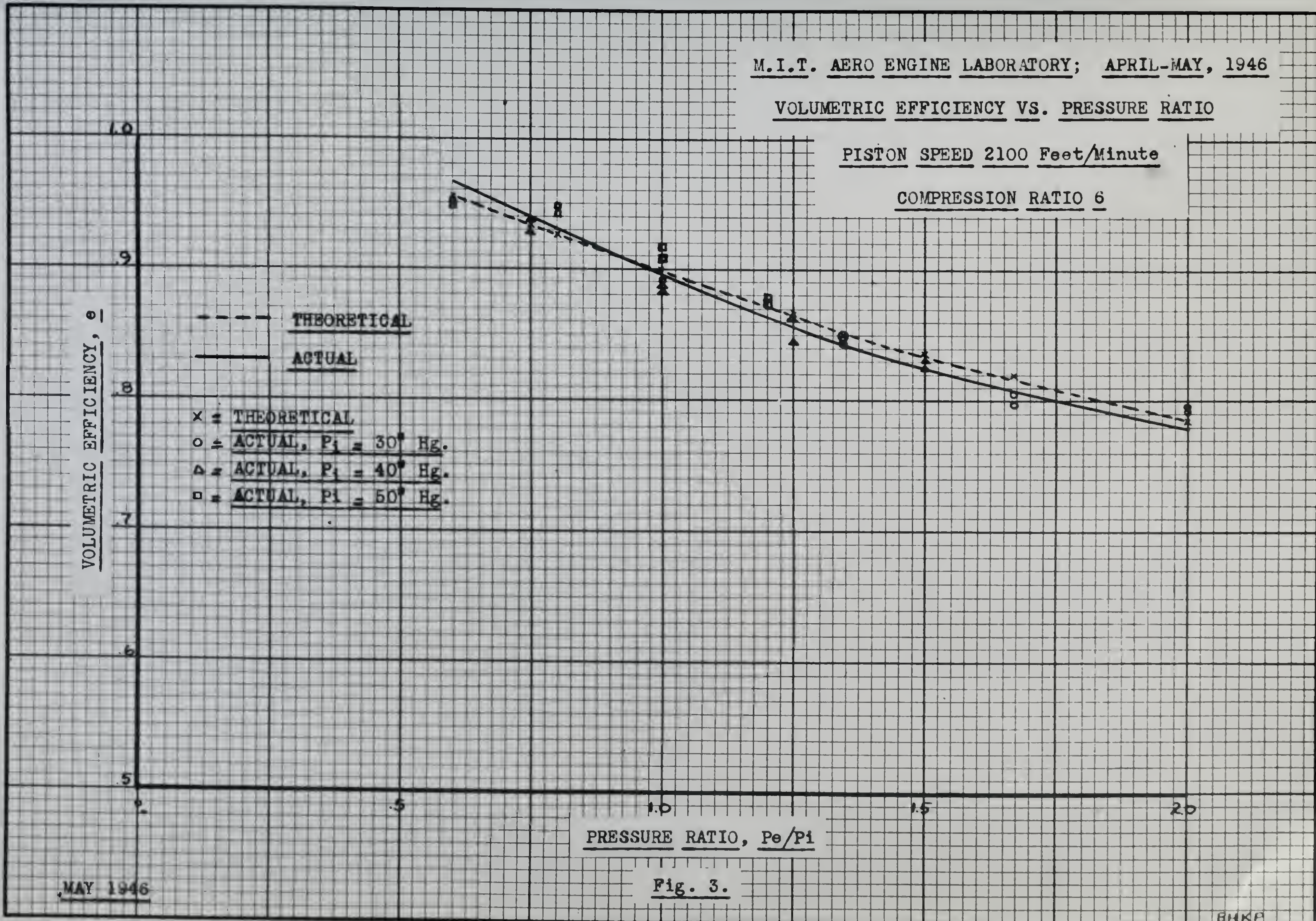
MAY 1946

BHKP



△ Single Plug, 38° Spark Advance
⊖ Dual Plug, 28° Spark Advance
Pi = 50" Hg.
Pi = 40" Hg.
Pi = 30" Hg.
1.0
0.9
0.8
0.7
0.6
0.5
VOLUMETRIC EFFICIENCY, e
30
40
50
60
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
VOLUMETRIC EFFICIENCY VS. EXHAUST PRESSURE
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

Fig. 2.



M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
VOLUMETRIC EFFICIENCY VS. PRESSURE RATIO
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
VOLUMETRIC EFFICIENCY, e
THEORETICAL
ACTUAL
x = THEORETICAL
o = ACTUAL, Pi = 30" Hg.
Δ = ACTUAL, Pi = 40" Hg.
□ = ACTUAL, Pi = 50" Hg.
1.0
.9
.8
.7
.6
.5
0
.5
1.0
1.5
2.0
PRESSURE RATIO, Pe/Pi
MAY 1946
BHKP

Fig. 3.

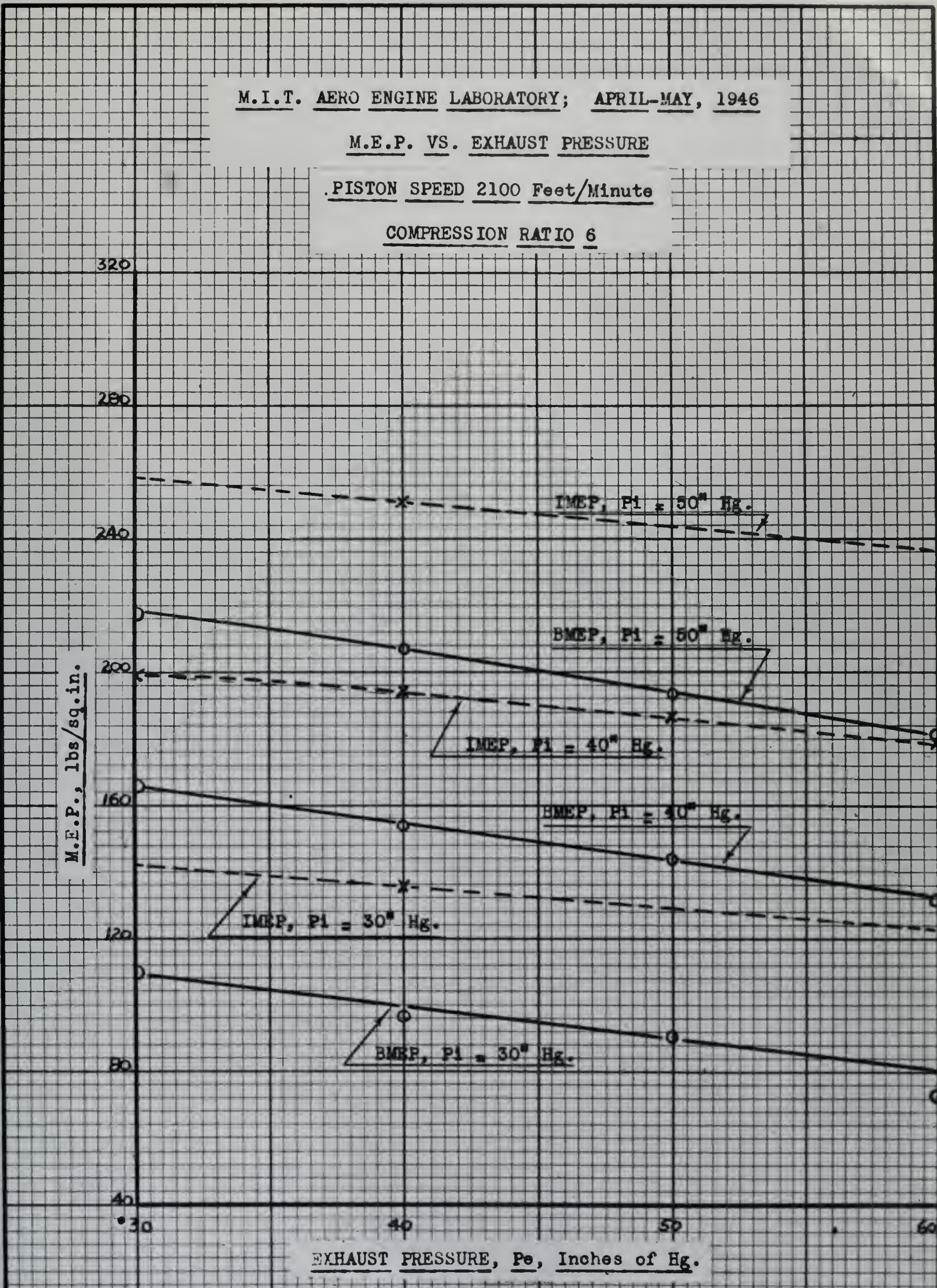

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
M.E.P. VS. EXHAUST PRESSURE
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
320
280
240
200
160
120
80
40
M.E.P., lbs/sq.in.
IMEP, Pi = 50" Hg.
BMEP, Pi = 50" Hg.
IMEP, Pi = 40" Hg.
BMEP, Pi = 40" Hg.
IMEP, Pi = 30" Hg.
BMEP, Pi = 30" Hg.
30
40
50
60
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

Fig. 4.

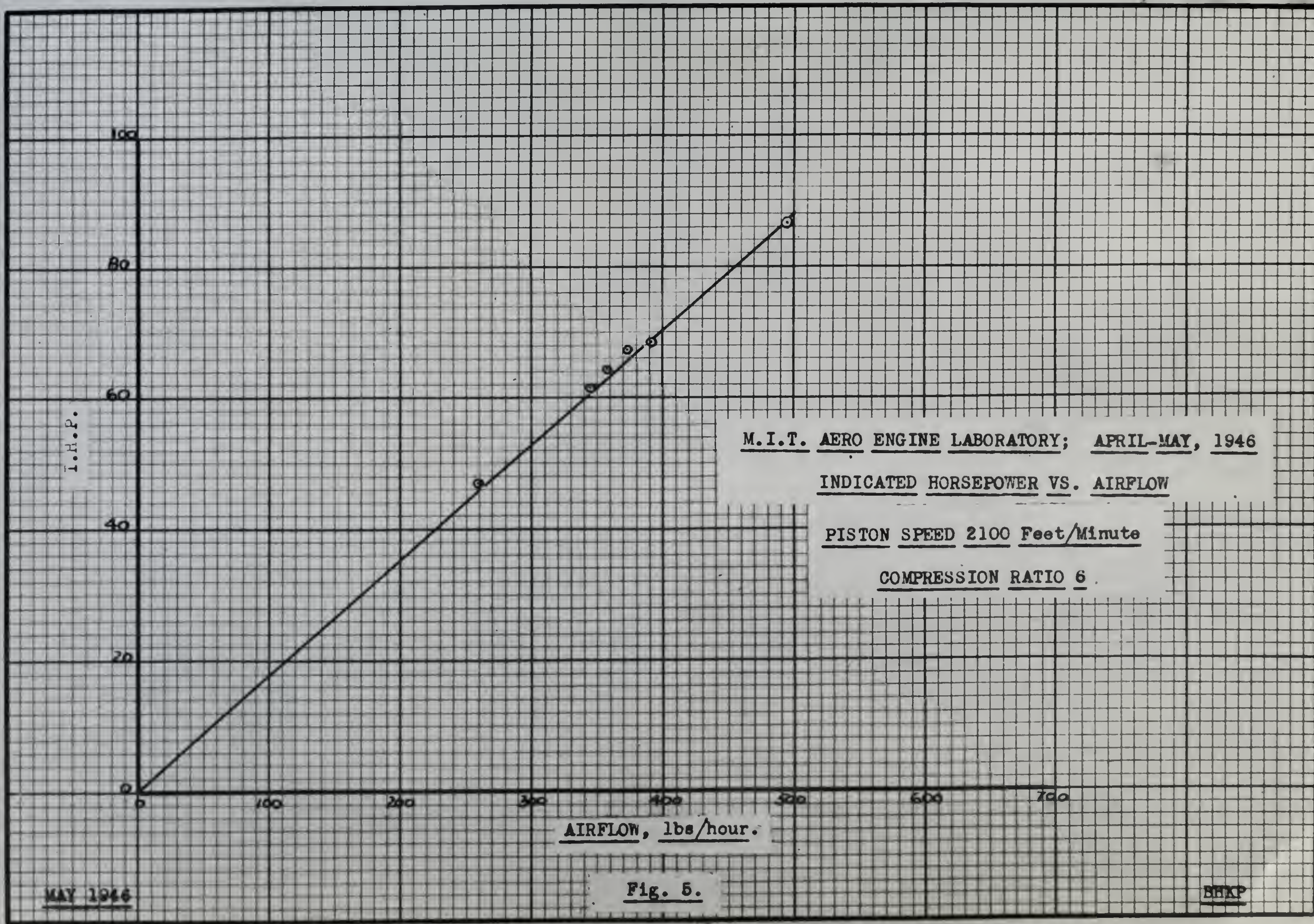

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
INDICATED HORSEPOWER VS. AIRFLOW
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
I.H.P.
100
80
60
40
20
0
0
100
200
300
400
500
600
700
AIRFLOW, lbs/hour.
MAY 1946
BHKP

Fig. 5.

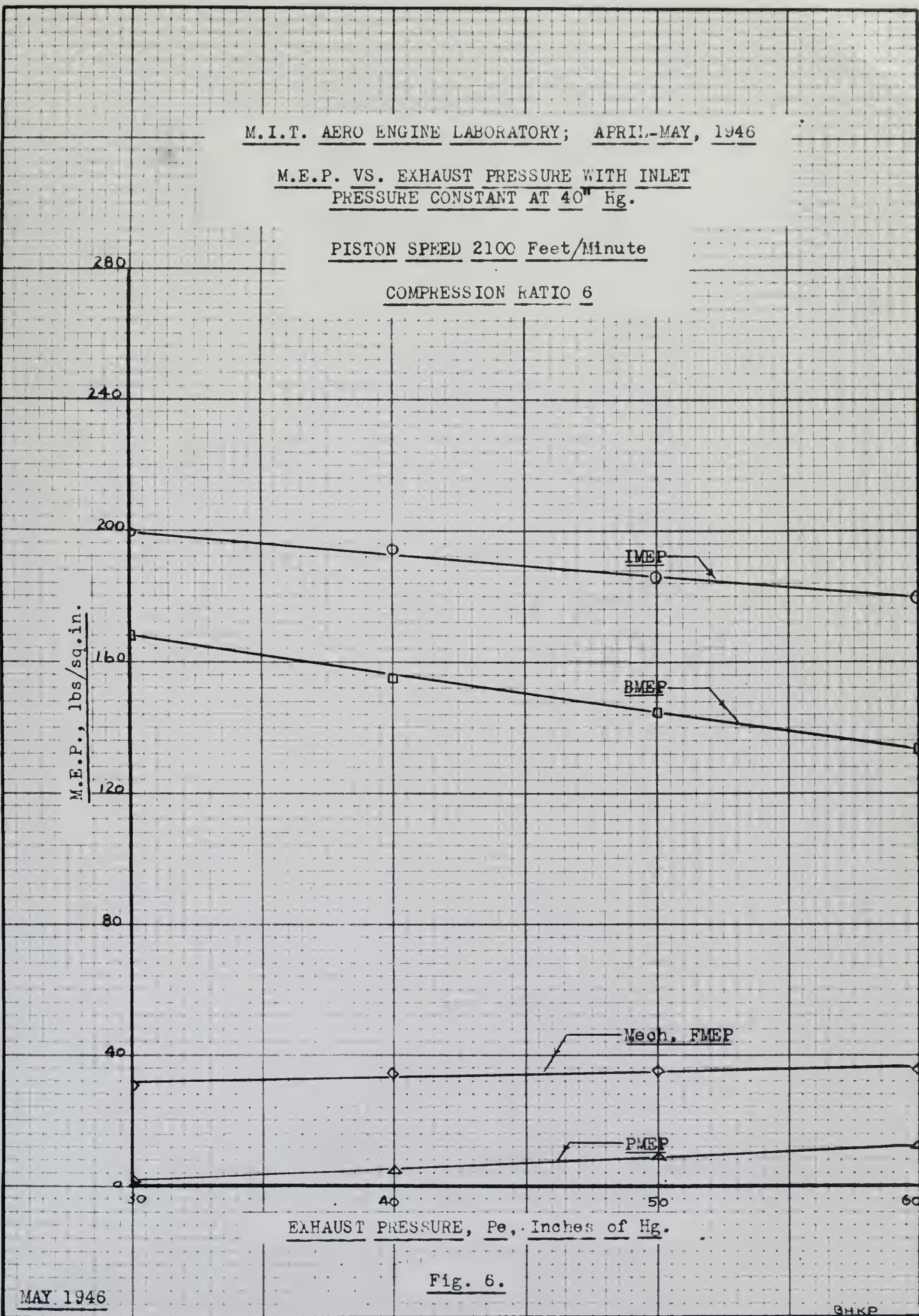

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
M.E.P. VS. EXHAUST PRESSURE WITH INLET PRESSURE CONSTANT AT 40" Hg.
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
M.E.P., lbs/sq.in.
280
240
200
160
120
80
40
0
IMEP
BMEP
Mech. FMEP
PMEP
30
40
50
60
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

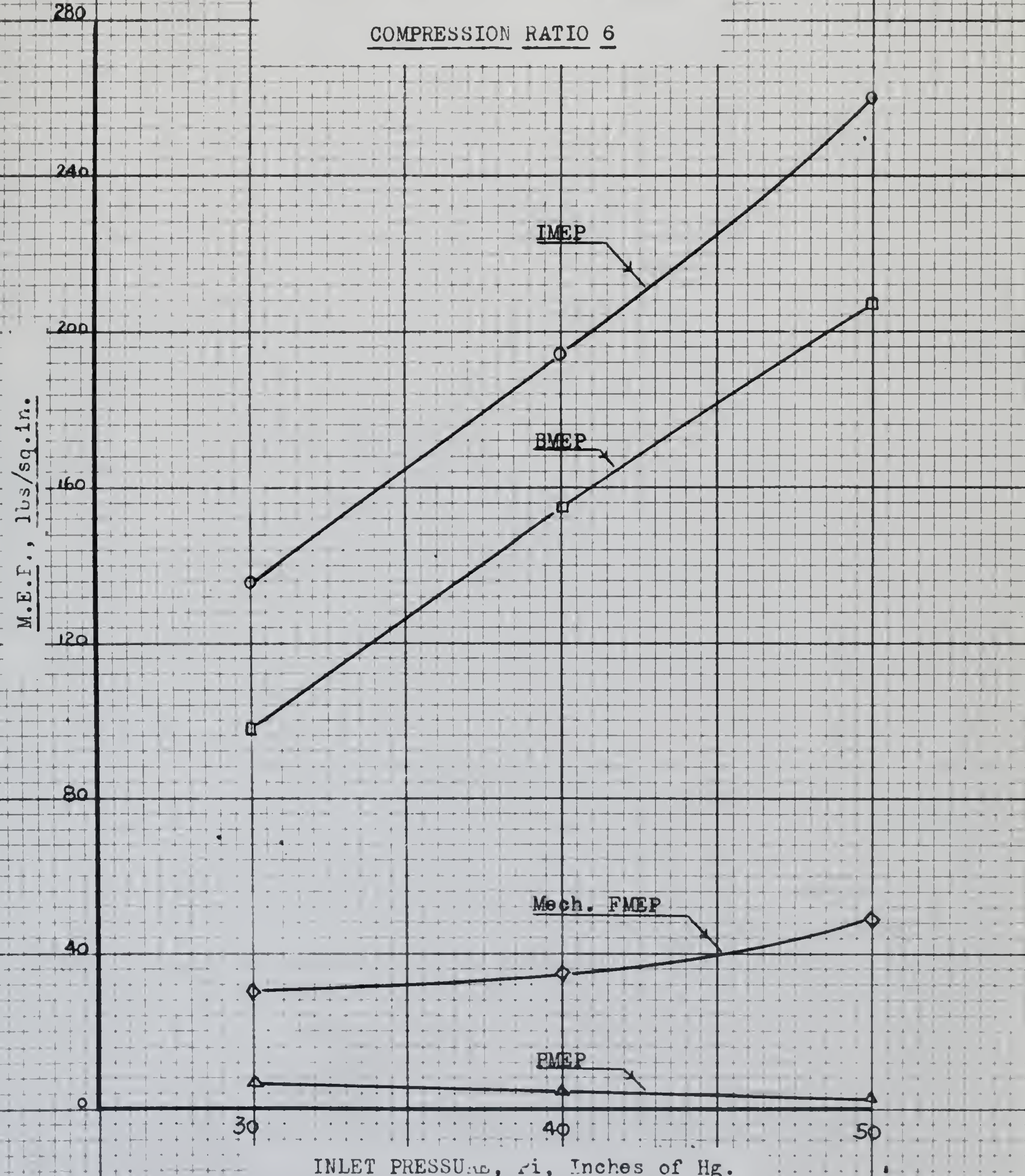
Fig. 6.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

M.E.P. VS. INLET PRESSURE WITH EXHAUST PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

Fig. 7.

BHKP

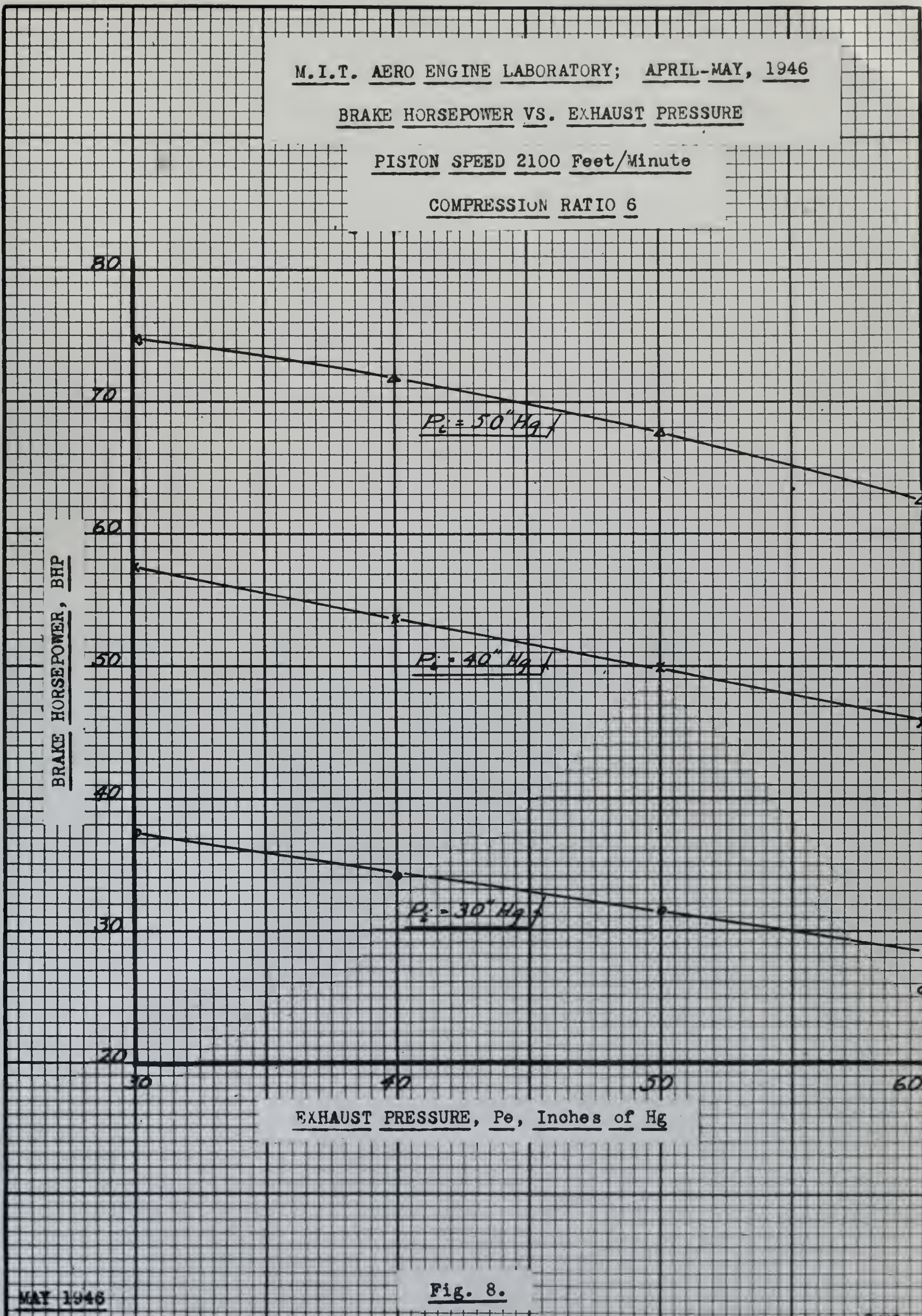
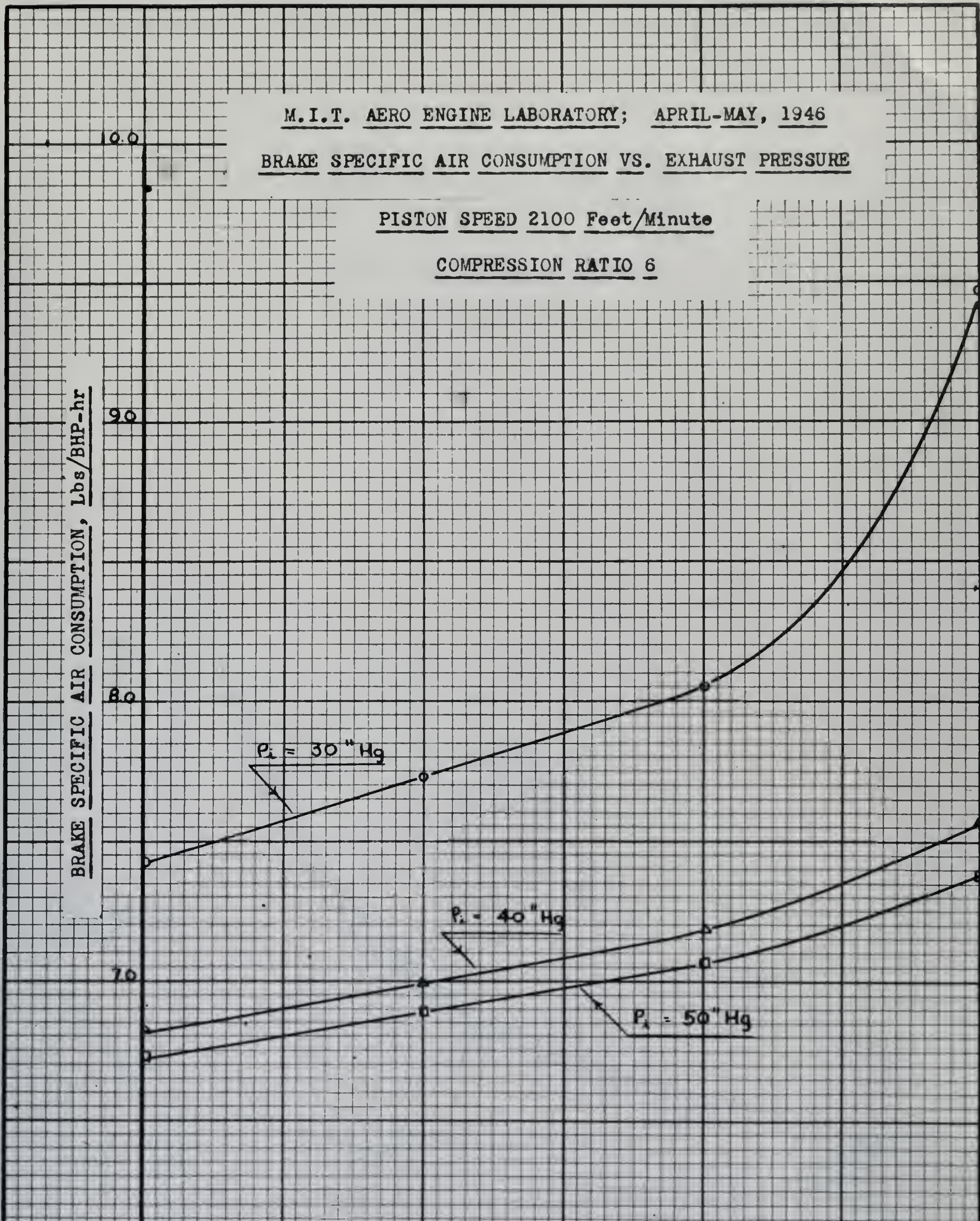

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
BRAKE HORSEPOWER VS. EXHAUST PRESSURE
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
BRAKE HORSEPOWER, BHP
80
70
60
50
40
30
20
$P_i = 50'' Hg$
$P_i = 40'' Hg$
$P_i = 30'' Hg$
30
40
50
60
EXHAUST PRESSURE, Pe, Inches of Hg
MAY 1946

Fig. 8.



M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
BRAKE SPECIFIC AIR CONSUMPTION VS. EXHAUST PRESSURE
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
BRAKE SPECIFIC AIR CONSUMPTION, Lbs/BHP-hr
10.0
9.0
8.0
7.0
6.0
P_i = 30" Hg
P_i = 40" Hg
P_i = 50" Hg
30
40
50
60
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946

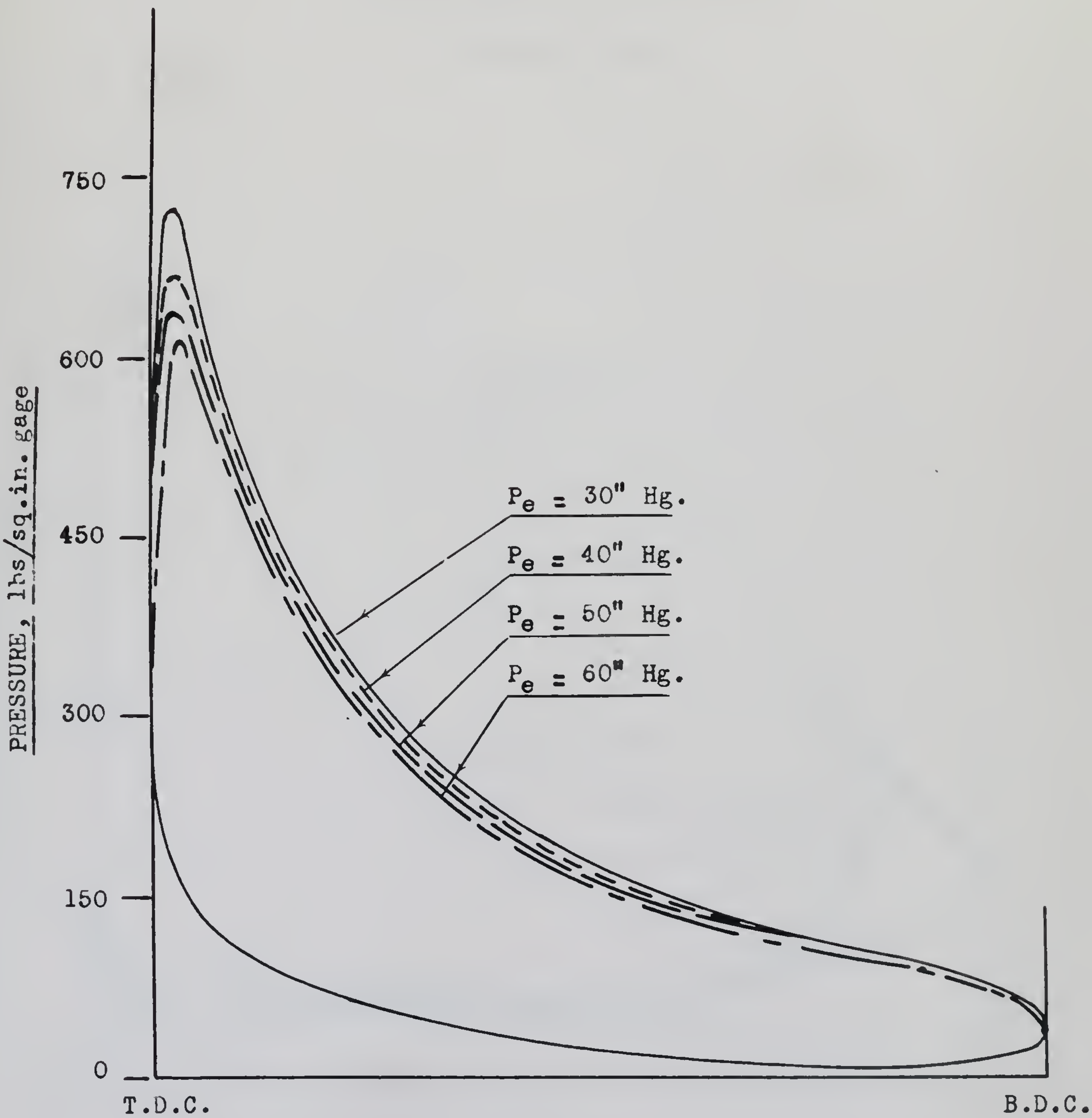
Fig. 9.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF EXHAUST PRESSURE ON
THE INDICATOR CARD WITHOUT PUMPING
LOOP WITH INLET PRESSURE CONSTANT
AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

BHKP

Fig. 10.

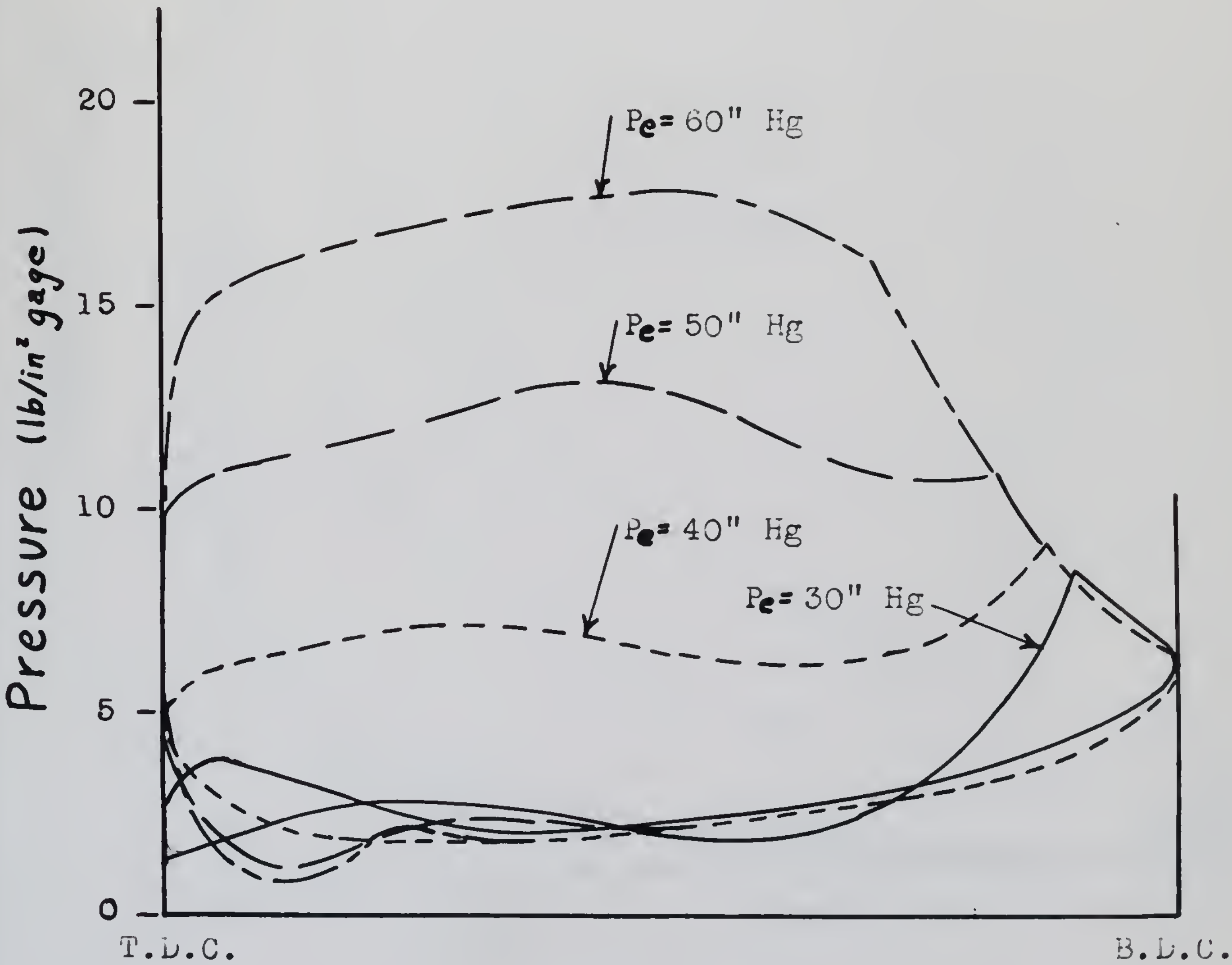
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF EXHAUST PRESSURE ON
THE PUMPING DIAGRAM WITH INLET
PRESSURE CONSTANT AT 40" Hg.

BORE STROKE ENGINE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



BHKP

Fig. 11.

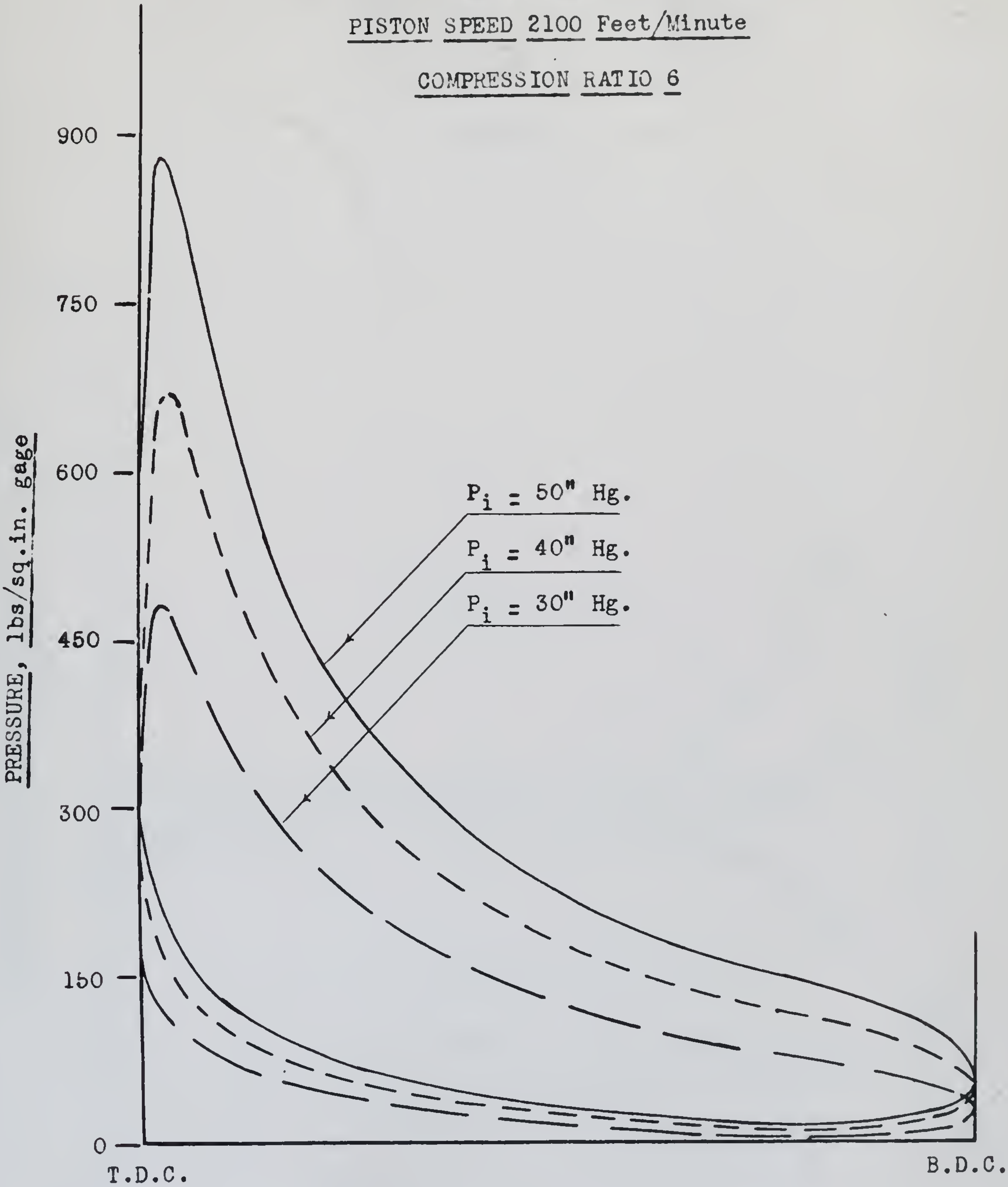
MAY 1946

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF INLET PRESSURE ON THE INDICATOR CARD WITHOUT PUMPING LOOP WITH EXHAUST PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

BHKP

Fig. 12.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF INLET PRESSURE ON THE
PUMPING DIAGRAM WITH EXHAUST
PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

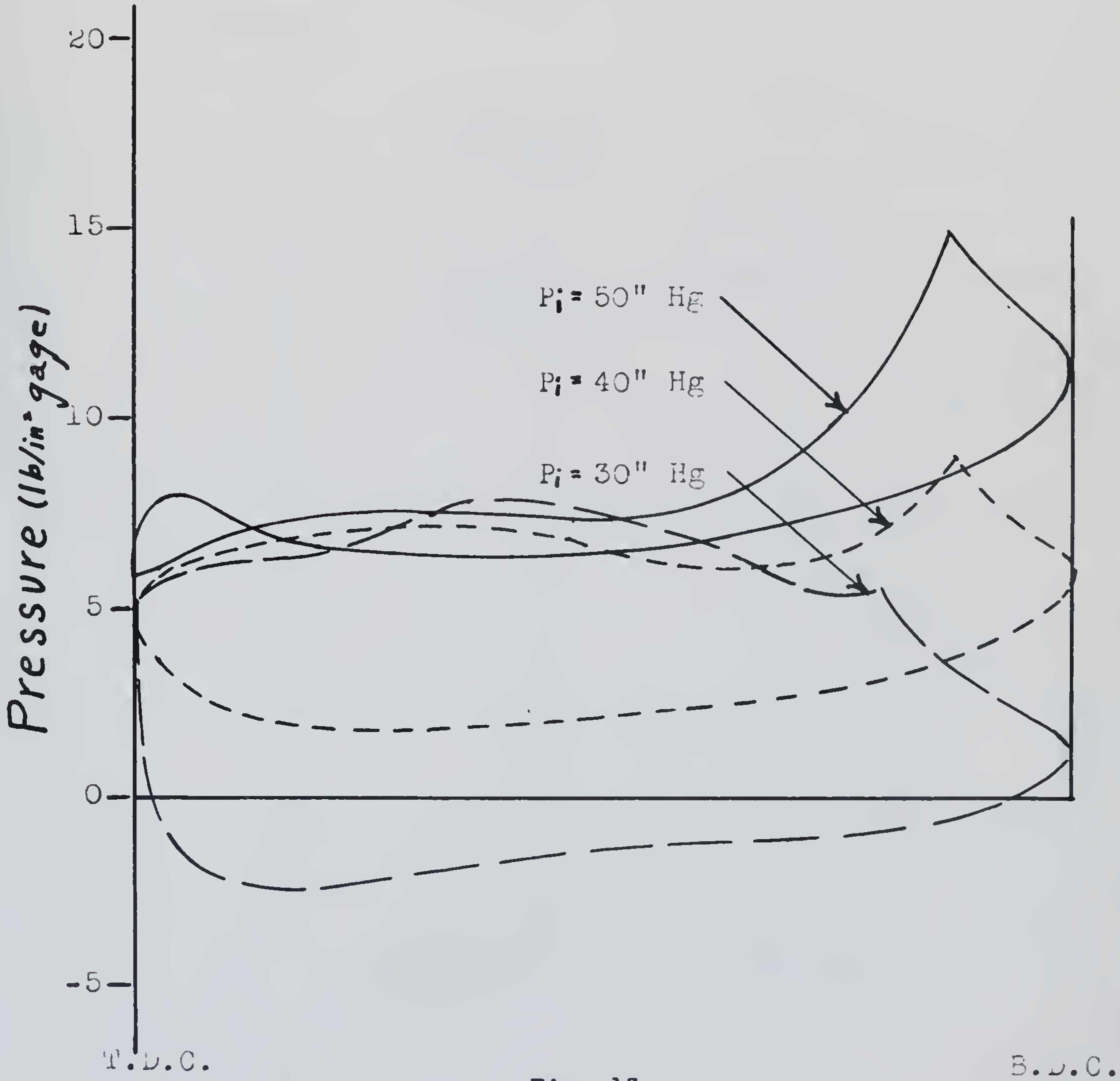


Fig. 13.

BHKP

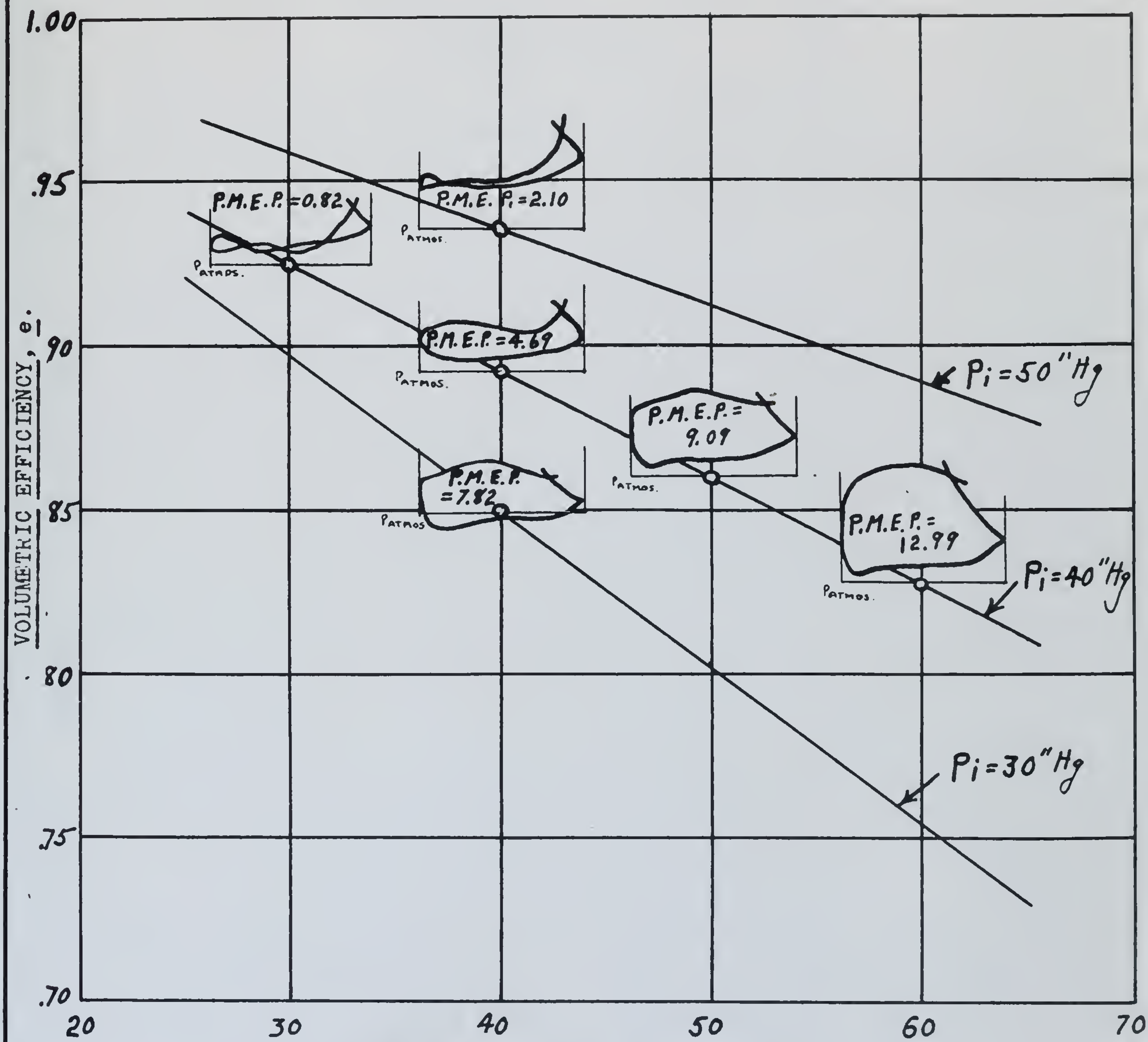
MAY 1946

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

RELATION OF PUMPING CYCLE TO
VOLUMETRIC EFFICIENCY

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



BHKP

Fig. 14.

MAY 1946

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

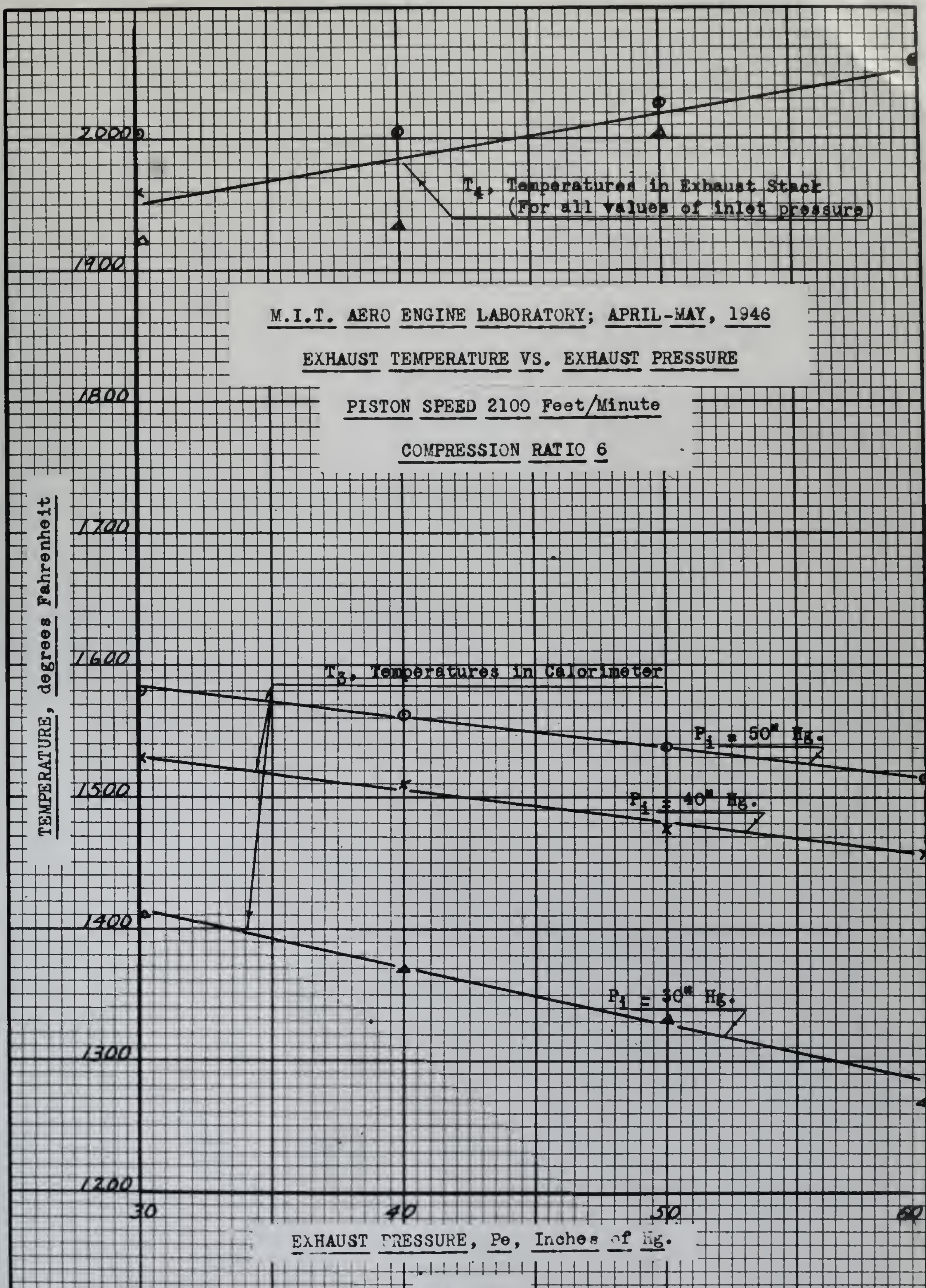
EXHAUST GAS TEMPERATURES VS. AMOUNT
OF OPENING OF BLEED VALVE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

Fig. 15.

MAY 1946



2000
1900
1800
1700
1600
1500
1400
1300
1200
30
40
50
60
T_4, Temperatures in Exhaust Stack (For all values of inlet pressure)
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
EXHAUST TEMPERATURE VS. EXHAUST PRESSURE
PISTON SPEED 2100 Feet/Minute
COMPRESSION RATIO 6
TEMPERATURE, degrees Fahrenheit
T_3, Temperatures in Calorimeter
P_i = 50" Hg.
P_i = 40" Hg.
P_i = 30" Hg.
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

Fig. 16.

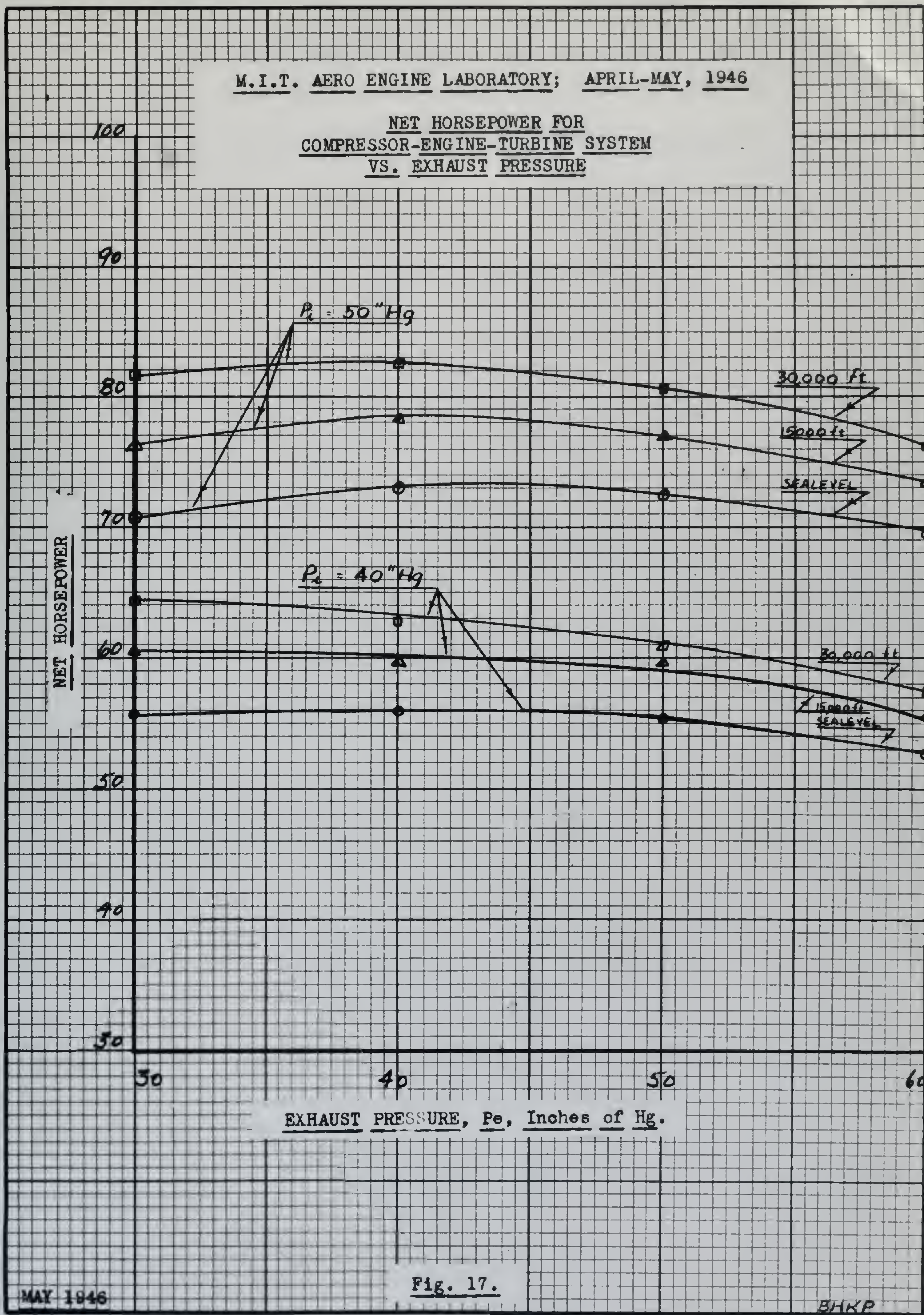

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
NET HORSEPOWER FOR
COMPRESSOR-ENGINE-TURBINE SYSTEM
VS. EXHAUST PRESSURE
100
90
80
70
60
50
40
30
NET HORSEPOWER
Pi = 50" Hg
Pi = 40" Hg
30,000 ft.
15000 ft
SEALEVEL
30,000 ft
15000 ft
SEALEVEL
30
40
50
60
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

Fig. 17.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

NET HORSEPOWER VS. RATIO OF EXHAUST PRESSURE/INLET PRESSURE FOR CET SYSTEM

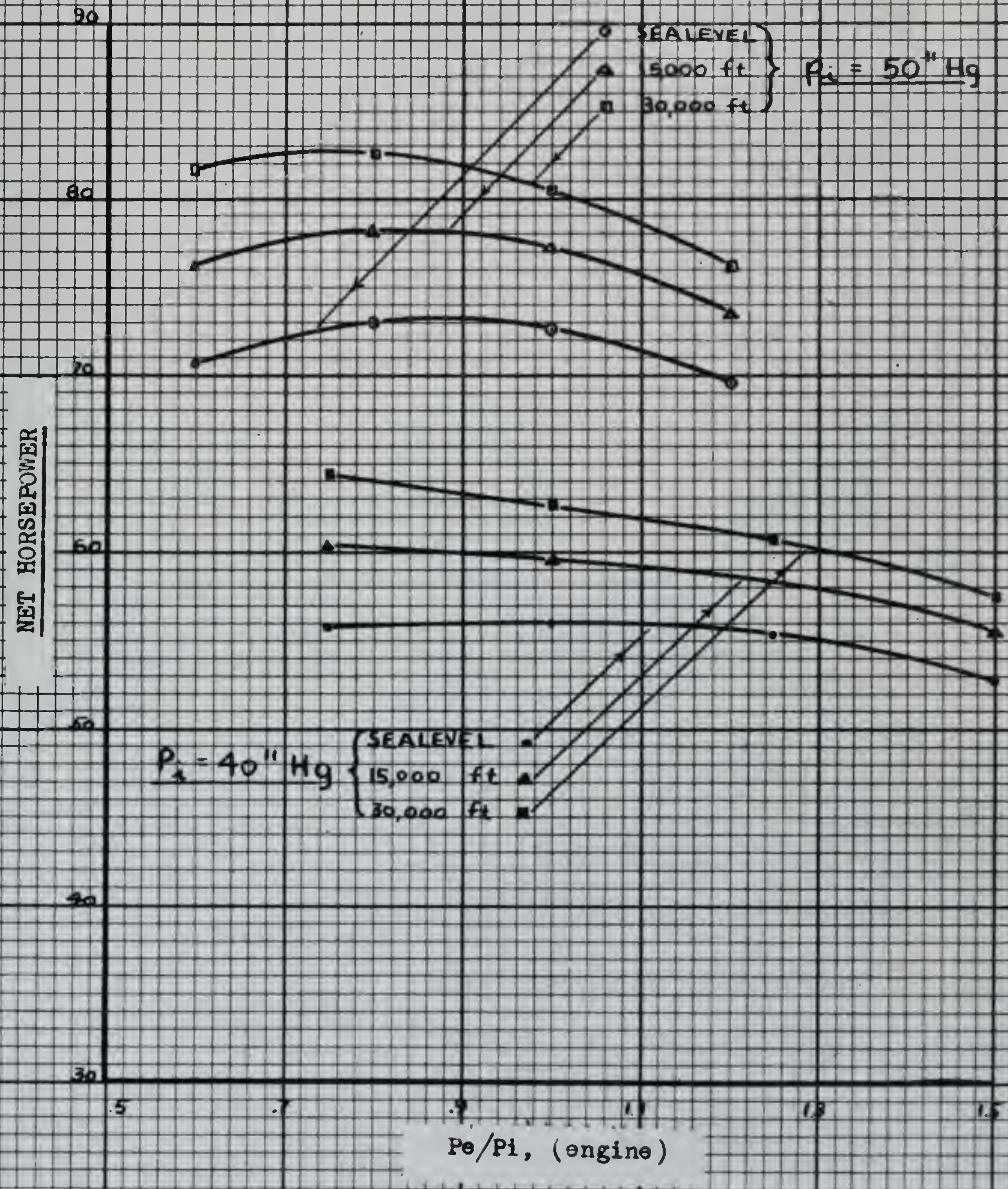


Fig. 18.

MAY 1946

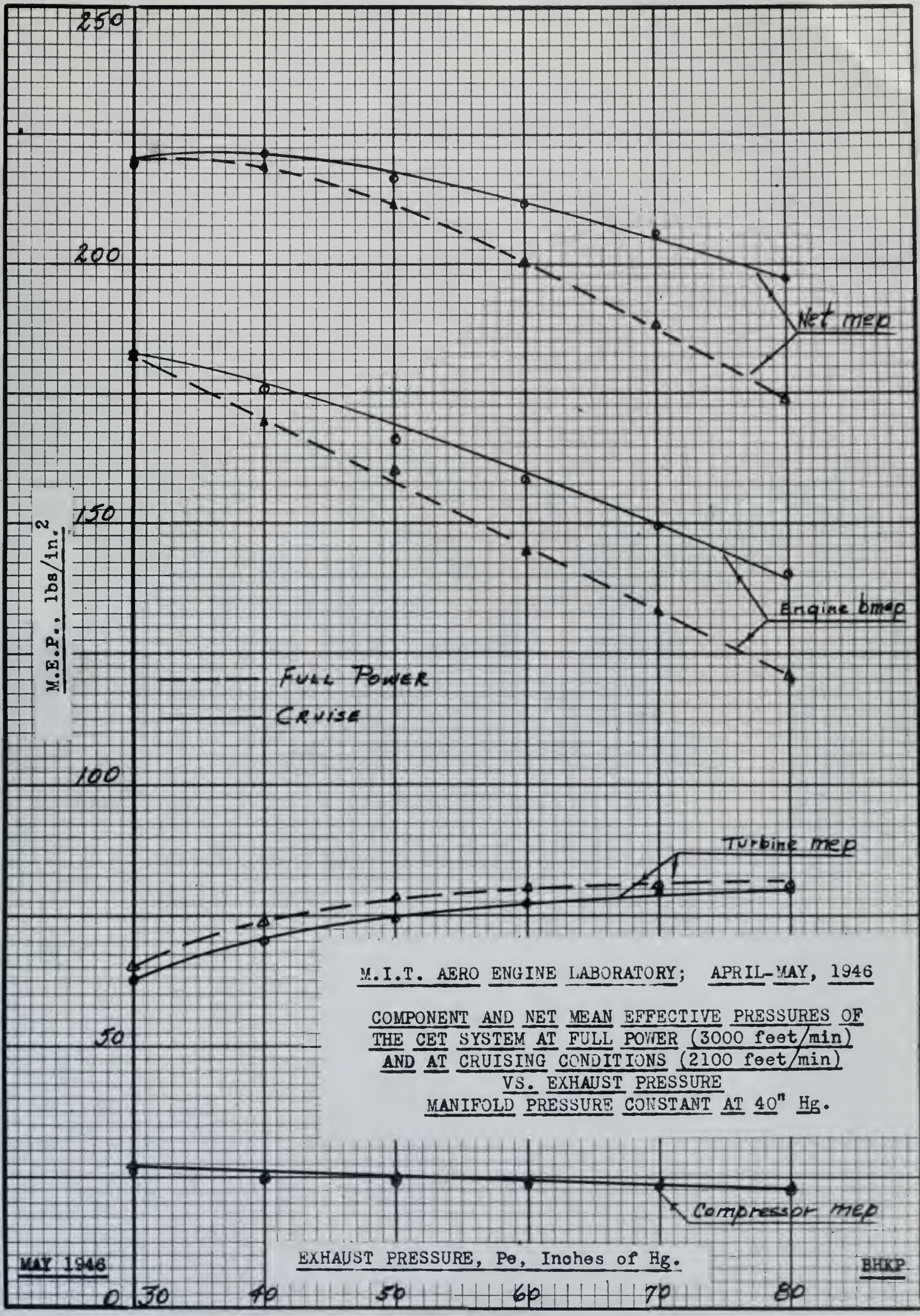
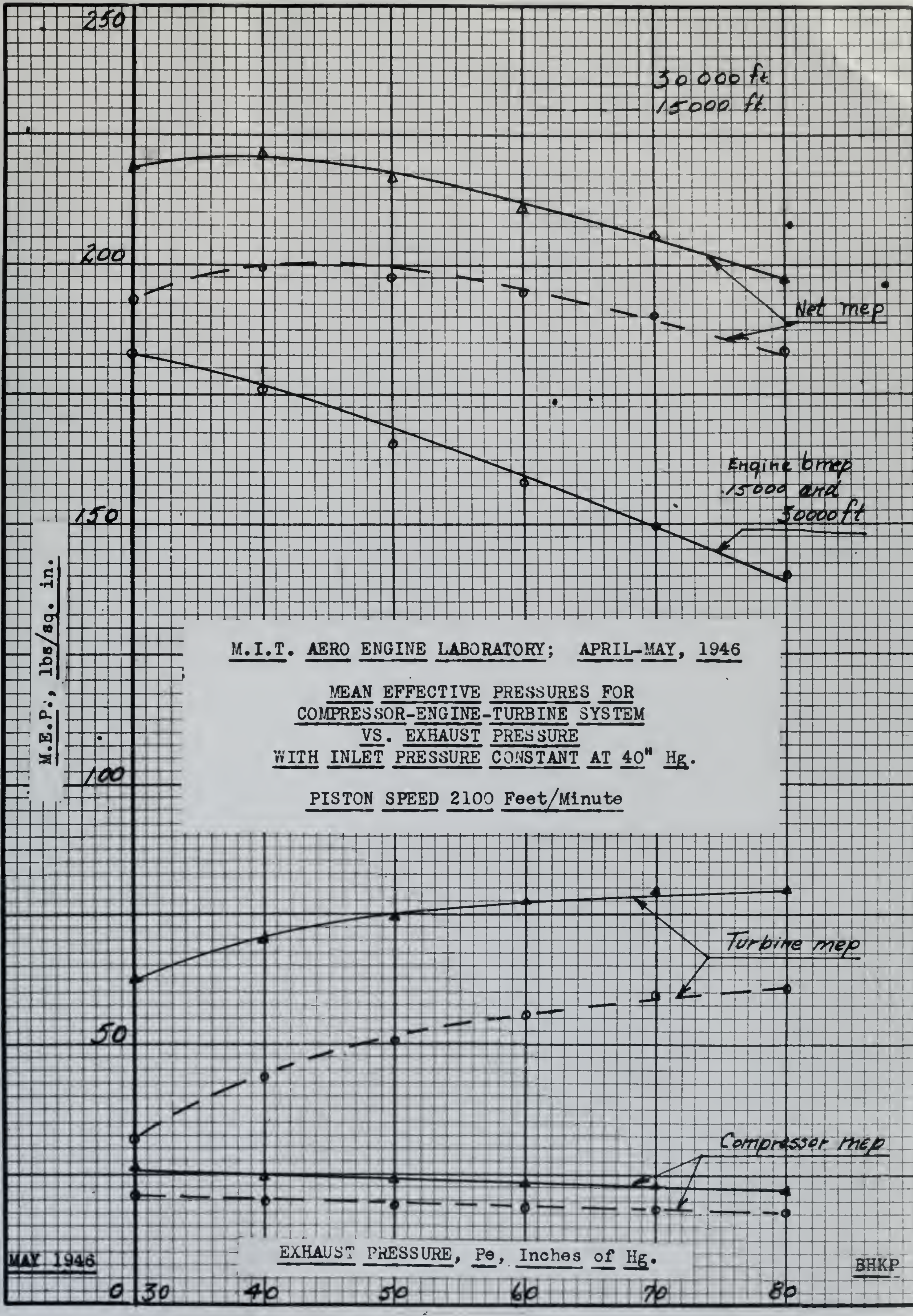
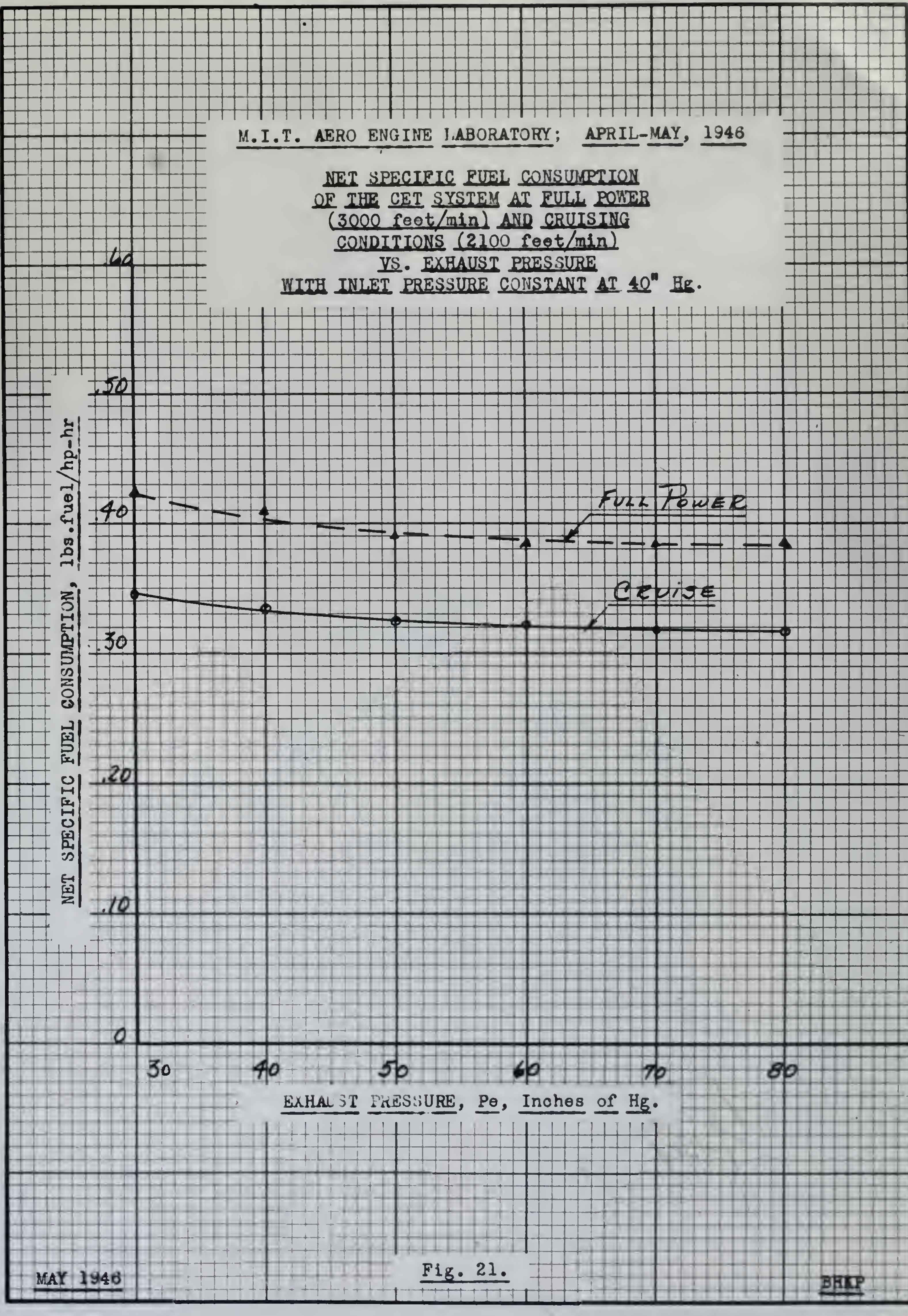

250
200
150
100
50
0
30
40
50
60
70
80
M.E.P., lbs/in.^{2}
Net mep
Engine bmep
Turbine mep
Compressor mep
FULL POWER
CRUISE
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
COMPONENT AND NET MEAN EFFECTIVE PRESSURES OF THE CET SYSTEM AT FULL POWER (3000 feet/min) AND AT CRUISING CONDITIONS (2100 feet/min) VS. EXHAUST PRESSURE
MANIFOLD PRESSURE CONSTANT AT 40" Hg.
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP

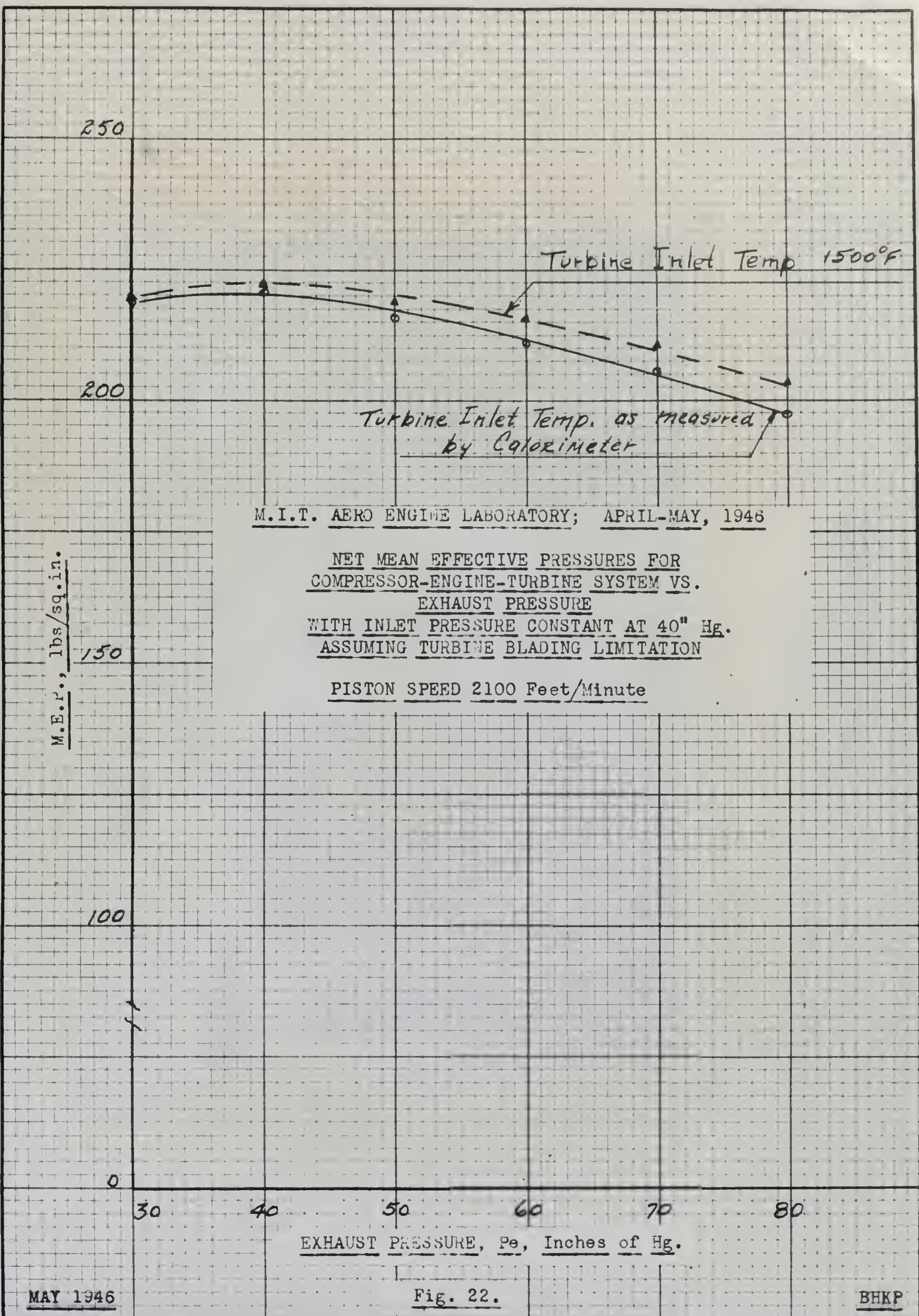
Fig. 19.



30 000 ft.
15000 ft.
Net mep
Engine bmep 15000 and 30000 ft
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
MEAN EFFECTIVE PRESSURES FOR
COMPRESSOR-ENGINE-TURBINE SYSTEM
VS. EXHAUST PRESSURE
WITH INLET PRESSURE CONSTANT AT 40" Hg.
PISTON SPEED 2100 Feet/Minute
M.E.P., lbs/sq. in.
250
200
150
100
50
0
Turbine mep
Compressor mep
EXHAUST PRESSURE, Pe, Inches of Hg.
30
40
50
60
70
80
MAY 1946
BHKP

Fig. 20.


M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
NET SPECIFIC FUEL CONSUMPTION
OF THE CET SYSTEM AT FULL POWER
(3000 feet/min) AND CRUISING
CONDITIONS (2100 feet/min)
VS. EXHAUST PRESSURE
WITH INLET PRESSURE CONSTANT AT 40" Hg.
NET SPECIFIC FUEL CONSUMPTION, lbs.fuel/hp-hr
.60
.50
.40
.30
.20
.10
0
FULL POWER
CRUISE
30
40
50
60
70
80
EXHAUST PRESSURE, Pe, Inches of Hg.
MAY 1946
BHKP
Fig. 21.



250
200
150
100
0
M.E.P., lbs/sq.in.
30
40
50
60
70
80
EXHAUST PRESSURE, Pe, Inches of Hg.
Turbine Inlet Temp 1500°F
Turbine Inlet Temp. as measured by Calorimeter
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946
NET MEAN EFFECTIVE PRESSURES FOR COMPRESSOR-ENGINE-TURBINE SYSTEM VS. EXHAUST PRESSURE WITH INLET PRESSURE CONSTANT AT 40" Hg. ASSUMING TURBINE BLADING LIMITATION
PISTON SPEED 2100 Feet/Minute
MAY 1946
BHKP

Fig. 22.

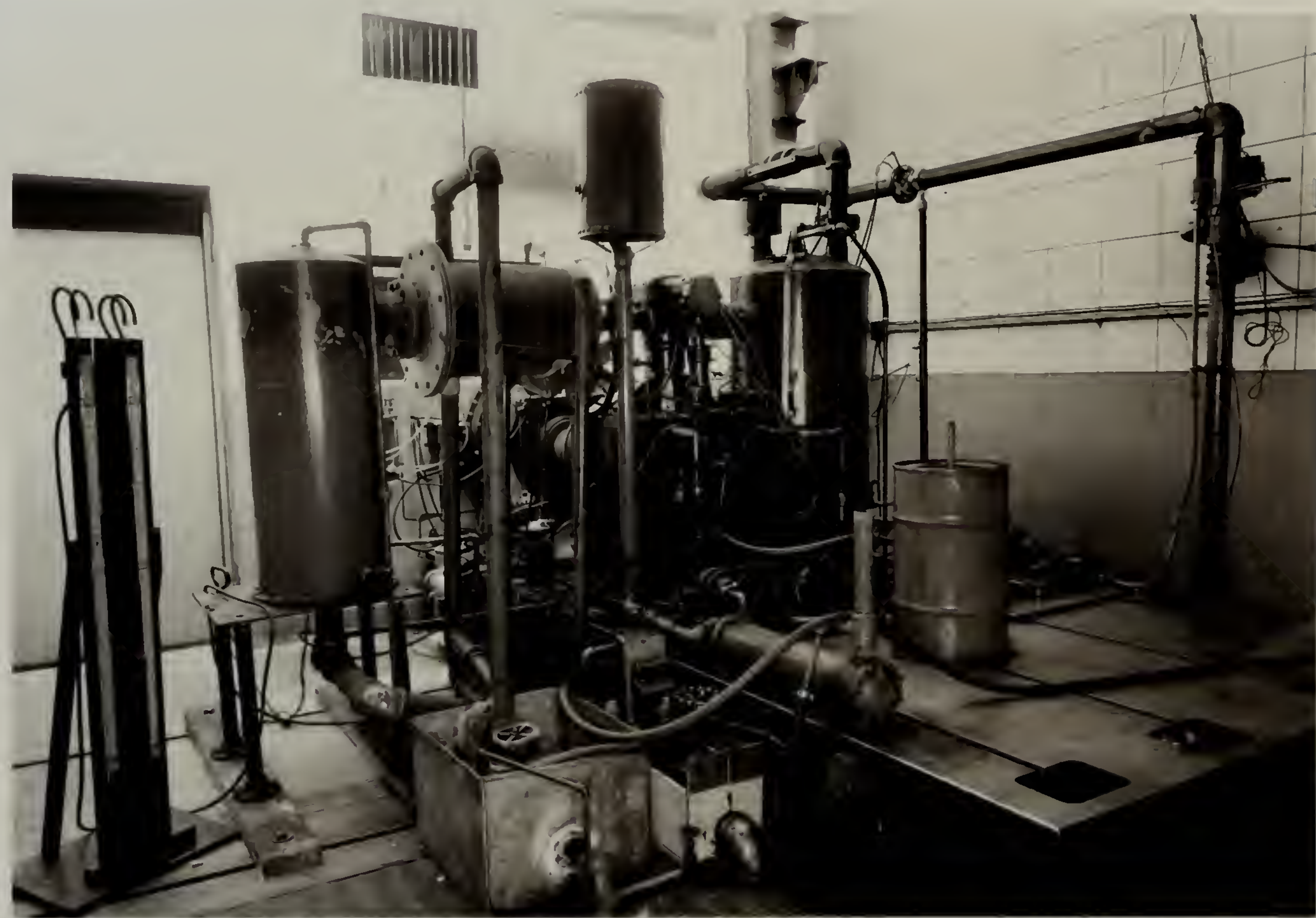


Fig. A.

ARRANGEMENT OF APPARATUS

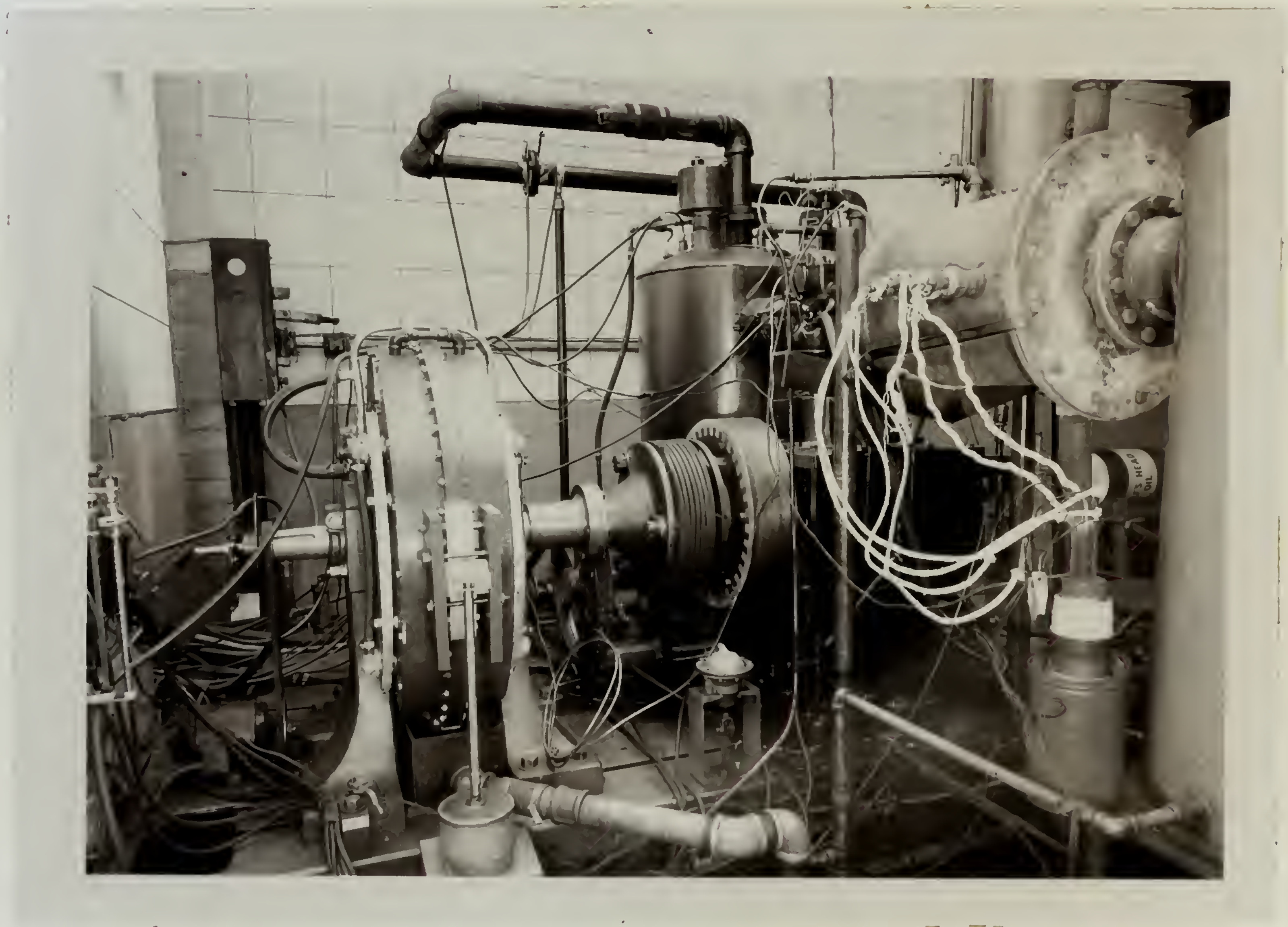


Fig. B

ARRANGEMENT OF APPARATUS

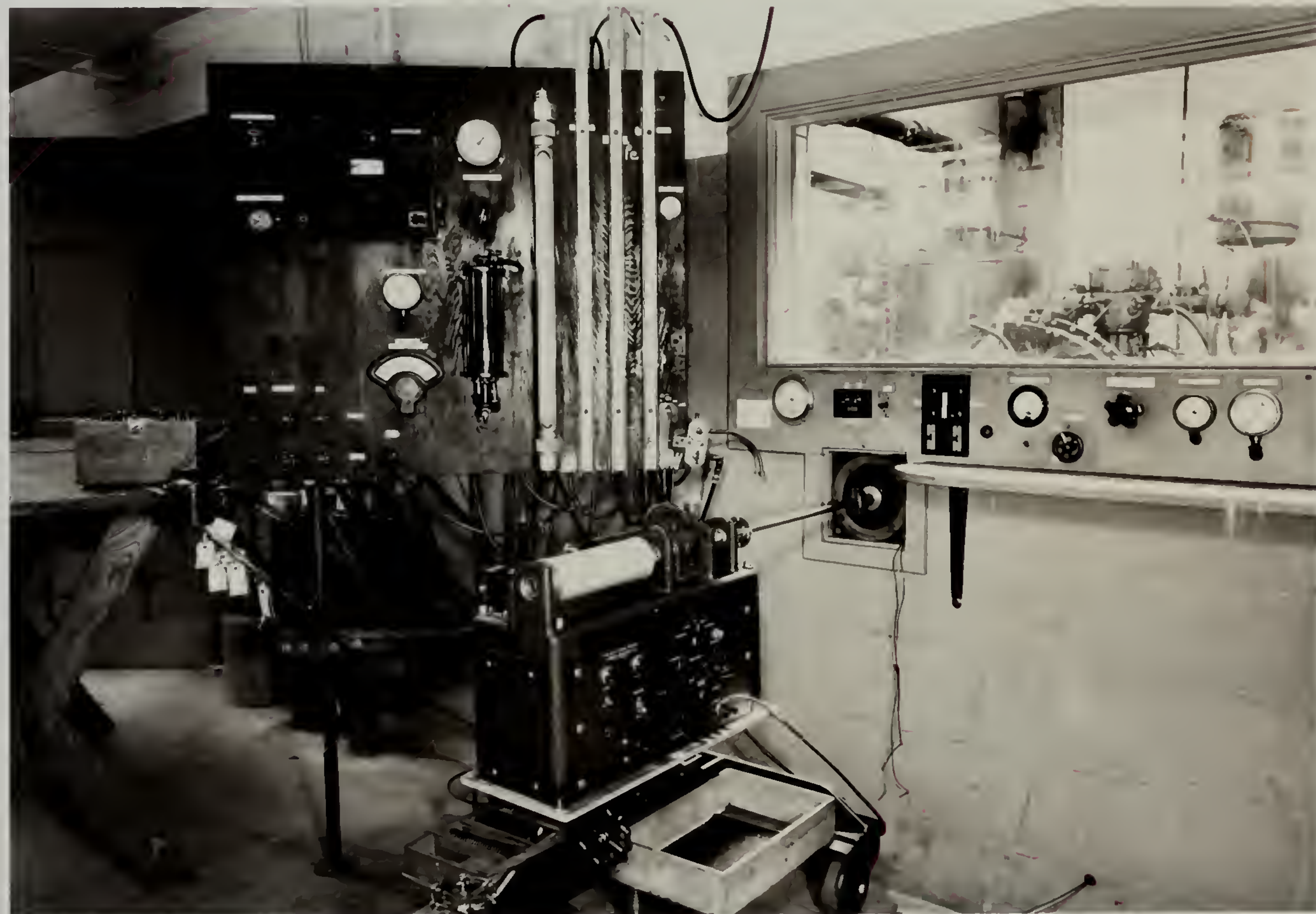


Fig. C

ARRANGEMENT OF APPARATUS

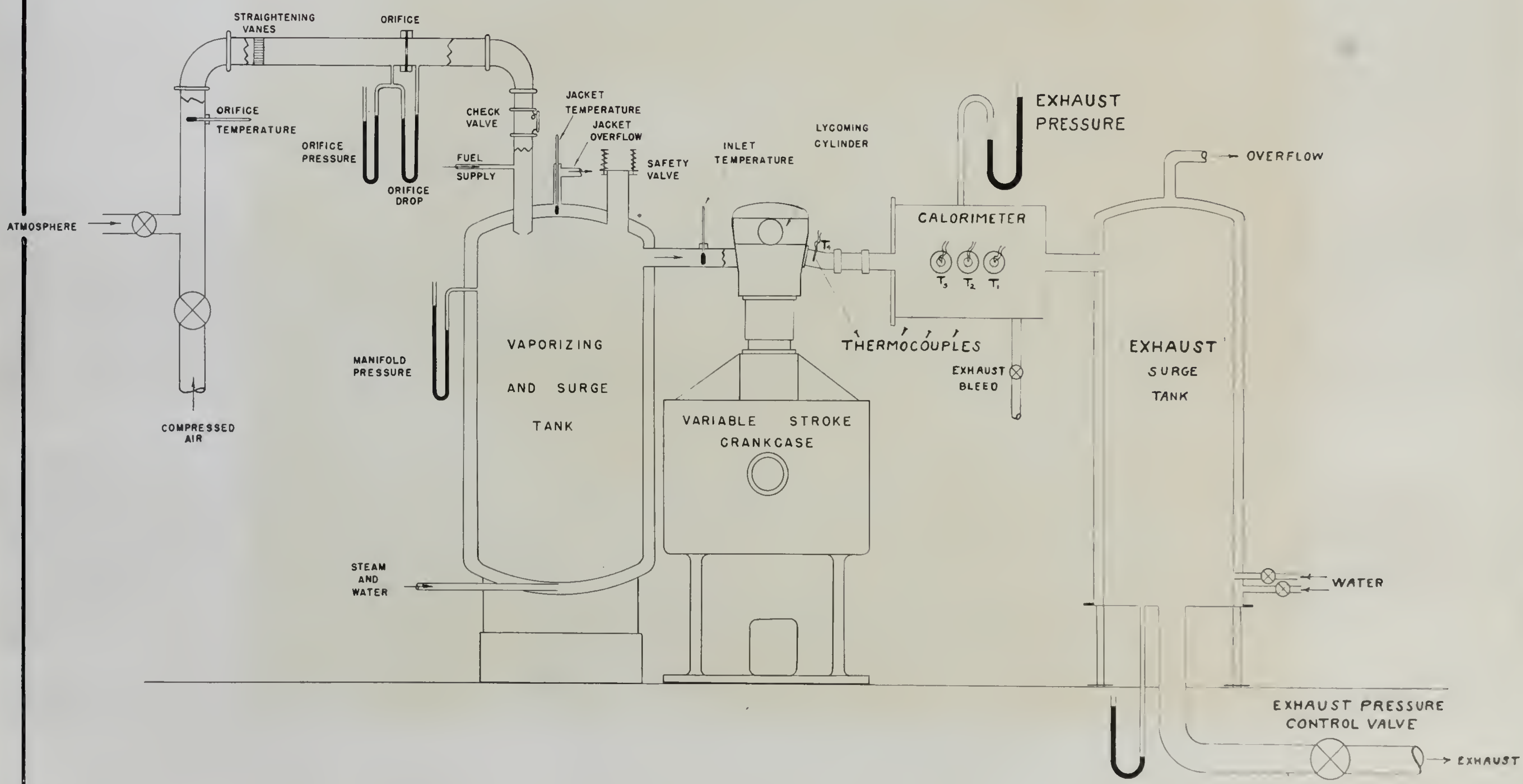


Fig. D

DIAGRAMATIC SKETCH OF ENGINE SET-UP

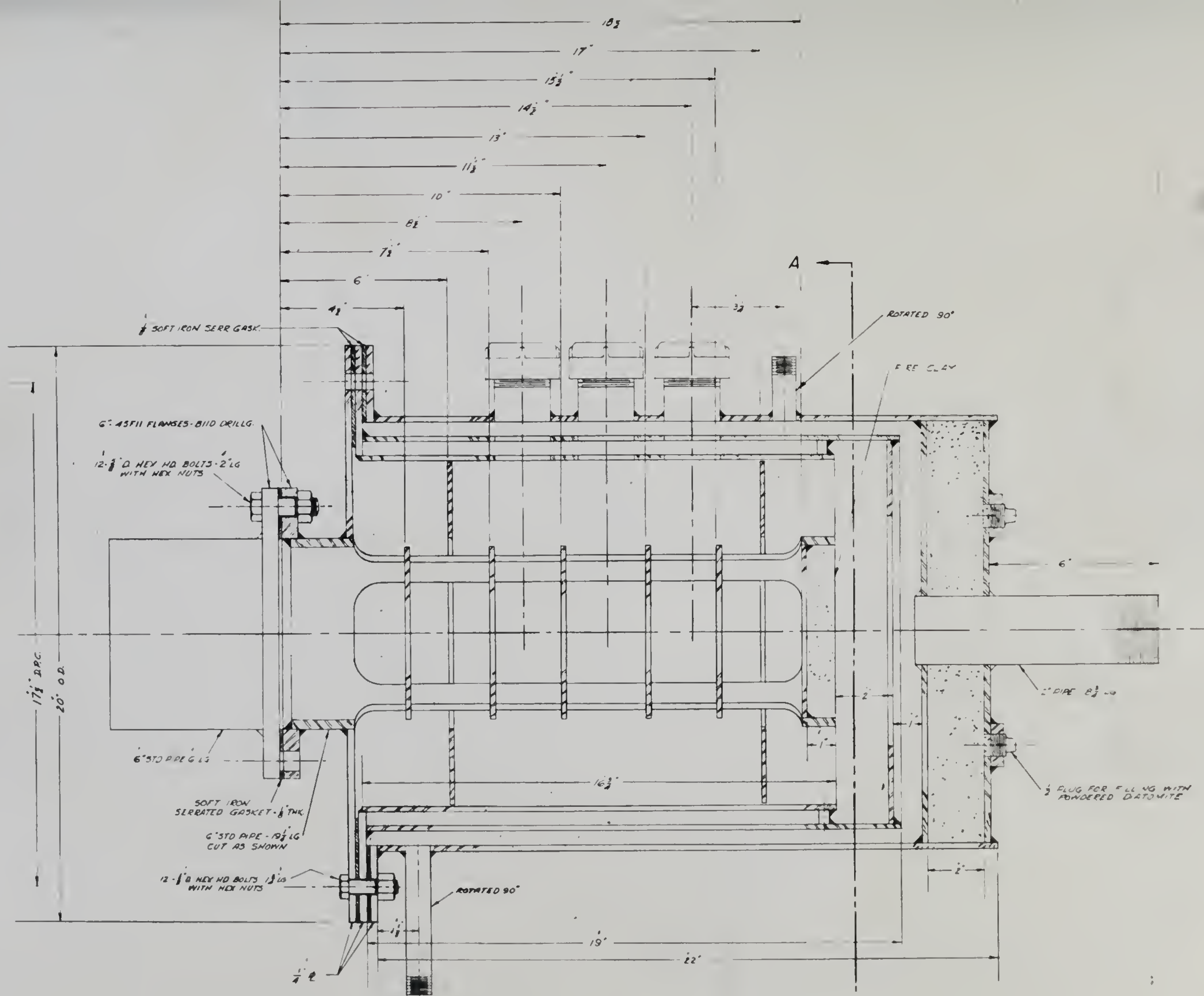


Fig. E

DIAGRAM OF EXHAUST CALORIMETER

VALVE TIMING DIAGRAM

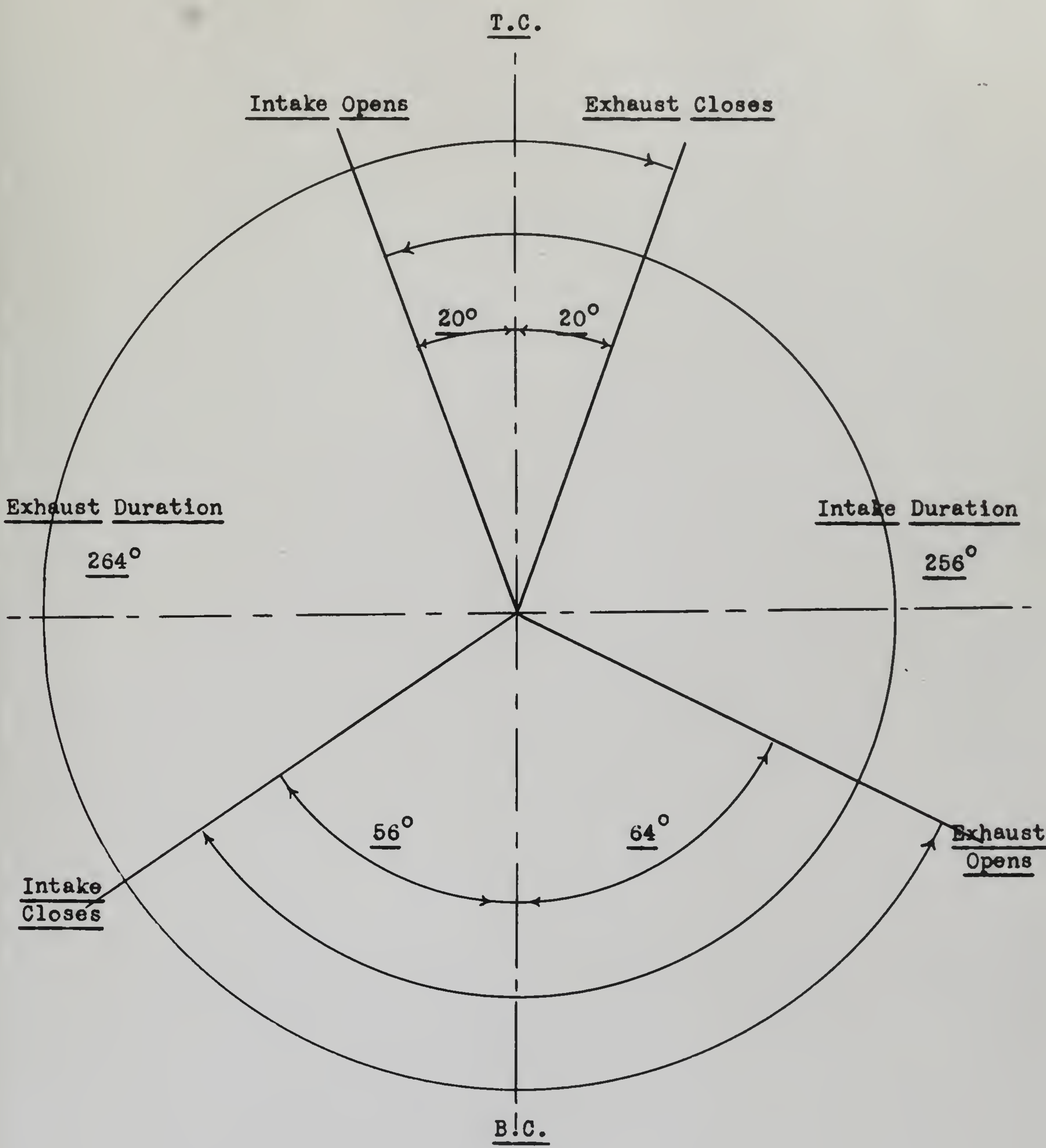


Fig. F

MAY 1946

BHKP

DATE DUE
